BACHELOR THESIS & COLLOQUIUM – ME 141502

Design and Simulation of Axial Turbine for Ocean Thermal Energy Conversion (OTEC)

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DOUBLE DEGREE PROGRAM OF
MARINE ENGINEERING DEPARTMENT
Faculty of Marine Technology
Institut Teknologi Sepuluh Nopember
Surabaya
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Desain dan Simulasi Turbin Aksial untuk Pembangkit Energi Energi Panas Laut (OTEC)

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2018

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APPROVAL FORM

Design and Simulation of Axial Turbine for Ocean Thermal Energy Conversion (OTEC)

BACHELOR THESIS & COLLOQUIUM

Asked to fulfill one of the requirements of obtaining a Double Degree of Bachelor Engineering

In

Study Field Marine Manufacturing and Design (MMD)

S-1 Double Degree Study Program Department of Marine Engineering

Faculty of Marine Technology

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ABSTRACT

Indonesia faces a decreasing in fossil energy reserves about 3% every year (Dewan Energi Nasional, 2015), and has not been matched by the discovery of new energy reserves. Therefore, it is necessary to increase the use of renewable Energy to meet energy needs. Renewable energy is energy derived from sustainable natural processes. Indonesia located on tropical area, it has a lot of potential ocean energy. OTEC (Ocean Thermal Energy Conversion) is one of many renewable energy sources from the ocean. OTEC or Ocean Thermal Energy Conversion is one of the latest technologies that used the temperature difference between deep and shallow seawater. OTEC system generally used ammonia (NH3) as working fluid. Ammonia is used because it has a relatively low boiling point compared to water. OTEC system consists of evaporators, turbines, generators, condensers, and pumps. In this research, the authors focused on the design of lab-scale OTEC turbines. 13 turbine models are designed and compared to get the highest efficiency and net power. The CFD method is used in the design and simulation. Based on the simulation results, calculations are done to known warm and cold seawater flow rate, pump power, and net power of the OTEC system. The highest efficiency and net power is a 3 stage 40 degree turbine with 351.37 kW generated power, and 65.02% efficiency. Lowest is single stage turbine with nett power 31.09 kW, and efficiency 45.85%.

Keywords: marine, renewable energy, turbine, ocean thermal, OTEC, CFD

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ABSTRAK

Indonesia menghadapi penurunan cadangan energi fosil sekitar 3% setiap tahun (Dewan Energi Nasional, 2015), dan belum diimbangi dengan penemuan cadangan energi baru. Oleh karena itu, perlu untuk meningkatkan penggunaan Energi terbarukan untuk memenuhi kebutuhan energi. Energi terbarukan adalah energi yang berasal dari proses alam berkelanjutan. Indonesia terletak di daerah tropis, memiliki banyak potensi energi laut. OTEC (Ocean Thermal Energy Conversion) adalah salah satu dari banyak sumber energi terbarukan dari lautan. OTEC atau Ocean Thermal Energy Conversion adalah salah satu teknologi terbaru yang menggunakan perbedaan suhu antara air laut dalam dan dangkal. Sistem OTEC umumnya menggunakan amonia (NH3) sebagai fluida kerja. Amonia digunakan karena memiliki titik didih yang relatif rendah dibandingkan dengan air. Sistem OTEC terdiri dari evaporator, turbin, generator, kondensor, dan pompa. Dalam penelitian ini, penulis memfokuskan pada desain turbin OTEC skala laboratorium. 13 model turbin dirancang dan dibandingkan untuk mendapatkan efisiensi dan daya bersih tertinggi. Metode CFD digunakan dalam desain dan simulasi. Berdasarkan hasil simulasi, perhitungan dilakukan untuk mengetahui debit air laut hangat dan dingin, daya pompa, dan daya bersih sistem OTEC. Efisiensi tertinggi dan daya bersih adalah turbin 3 stage 40 derajat dengan 350,91 kW daya yang dihasilkan, dan efisiensi 65,02%. Terendah adalah turbin satu tahap dengan daya nett 30,87 kW, dan efisiensi 45,85%.

Keywords: laut, energi terbarukan, turbin, panas laut, OTEC, CFD

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PREFACE

Alhamdulillahirabbil ‘alamin, huge thanks to Allah SWT the God Almighty for giving intelligent, strength, health and favours so the author can finish this bachelor thesis.

This bachelor thesis aims to know how to design Otec turbine, and the performance of its models. The author also would express his immeasureable appreciation and deepest gratitudefor those who helpedin completing this bachelor thesis:

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The author realizes that this thesis remains far away from perfect. Therefore, every constructive suggestions and idea from all parties are highly expected by the author to improve this bachelor thesis in future. Hopefully, this bachelor thesis can be advantages for all of us, particularry the readers.

Surabaya, July 2018

Author

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CHAPTER I
INTRODUCTION

1.1 Background

Energy has an important role in the achievement of social, economic and environmental goals for sustainable development, and national economic activities. Energy use in Indonesia is increasing rapidly in line with economic growth and population growth. Dependence on fossil energy, especially petroleum in the fulfilment of domestic consumption is still high, about 96% (petroleum 48%, gas 18%, and coal 30%) of total national energy consumption (National Energy Council, 2014). Indonesia faces a decreasing in fossil energy reserves about 3% every year (Dewan Energi Nasional, 2015), and has not been matched by the discovery of new energy reserves. Therefore, it is necessary to increase the use of renewable Energy to meet energy needs. Renewable energy is energy derived from sustainable natural processes.

Studies and projects on renewable energy sources have been actively conducted around the world to solve the challenge of energy supply and concomitant environmental issues. Among renewable energy sources, ocean energy has been used in various approaches and is known as an eco-friendly one. Indonesia located on tropical area, it has a lot of potency in ocean energy. OTEC (Ocean Thermal Energy Conversion) is one of many renewable energy sources from the ocean. As part of ocean energy development, OTEC (Ocean Thermal Energy Conversion) system has been applied in the United States, Europe, Japan, and other developed countries.

OTEC or Ocean Thermal Energy Conversion is one of the latest technologies that used the temperature difference between deep and shallow seawater that drive generators to produce electrical energy. The sunlight that falls on the oceans is so strongly absorbed by the water that effectively all of its energy is captured within a shallow "mixed layer" at the surface, 35 to 100 m thick, where wind and wave actions cause the temperature and salinity to be nearly uniform. In the tropical oceans between approximately 15° north and 15° south latitude, the heat absorbed from the sun warms the water in the mixed layer to a value near 28°C that is nearly constant day and night and from month to month. The annual average temperature of the mixed layer throughout the region varies from about 27°C to about 29°C. Beneath the mixed layer, the water becomes colder as depth increases until at 800 to 1000 m (2500 to 3300 ft), a temperature of 4.4°C (Avery & Wu, 1995). This temperature does not change dramatically throughout the year, with varying degrees due to weather and seasonal changes, and the temperature difference between day and night turns only has an effect of about 1°C (Soesilo, 2017).

Ocean Thermal Energy Conversion generating systems can be classified into three categories: open cycles, closed cycles, and hybrid cycles. OTEC closed cycle generally uses ammonia (NH3) as working fluid. Ammonia is used because it has a relatively low boiling point compared to water. OTEC system consists of evaporators, turbines, generators, condensers, and pumps. In this research, the authors focused on the design of lab-scale OTEC turbines to get the highest efficiency and net power.

1.2 Research Problem

Based on background mentioned above, it can be concluded some problems of this final project are:

1. How to design a steam turbine for OTEC System?
2. How is the performance of the OTEC turbine that has been simulated by CFD software?

1.3 Research Limitation

The limitations of the problems in this research are:

1. Focused only on design of OTEC turbine,
2. System are steady state,
3. Adiabatic,
4. RPM of turbine is 3000,
5. Radius of hub and shroud determined 0.06 m and 0.1 m,
6. This research used CFD software to simulated the turbine,
7. Does not include metallurgical analysis,
8. Does not include financial analysis.

1.4 Research Objectives

Based on problems mention above, the objectives of this research are:

1. To know how to design a steam turbine for OTEC System.
2. To know the performance of OTEC turbine that has been simulated by CFD software

1.5 Research Benefit

This research is expected to give benefits to several parties. Benefits that can be obtained from this research are:

1. Increase the knowledge of the author and readers about the potential of the ocean as renewable energy source.
2. Provide Information to develop OTEC as a renewable, and environmentally friendly power plant.

CHAPTER II
LITERATURE STUDY

2.1 Preliminary

Indonesia has a lot of energy resources, both in the fossil resources, and renewable natural resources. In renewable natural resources, Indonesia has excellent potential, one of them is maritime sector. *Dewan Energi Nasional* (English: National Energy Council) has mapped the potential of ocean's renewable energy as shown in **Table 1.**

Table 1. Ocean's Renewable Energy Potential in Indonesia
(Source: Dewan Energi Nasional, 2014)

Type	Theoritical Resource (MW)	Technical Resource (MW)	Practical Resource (MW)
Ocean Thermal	4.247.389	136.669	41.001
Ocean and Tidal Current	287.822	71.955	17.989
Ocean Wave	141.472	7.985	1.995
Total	**4.676.683**	**216.609**	**60.985**

From the table above, it can be concluded that Indonesia has great theoretical potential of ocean energy, especially on ocean thermal energy. However, it differs considerably in technical and practical potential. It is happens because of the limitations of technology used to develop ocean energy.

Indonesian government through Law no. 30 / 2007 on Energy and Law no. 17 / 2007 on the National Long-Term Development Plan (Bahasa Indonesia: Rencana Pembangunan Jangka Panjang Nasional) has supported the development of ocean energy. But in fact, the road map of ocean energy development and the National Electricity General Plan (Bahasa Indonesia: Rencana Umum Kelistrikan Nasional) has not accommodated the utilization of sea energy yet. This situation is caused by several things, including the unavailability of information about ocean energy that can economically be utilized for power generation, and technological limitations to support sea energy. Therefore, further research is needed on the development of technology used to support ocean energy.

Indonesia located on tropical area. It has a lot of potencial ocean energy, especially for ocean thermal energy conversion (OTEC). In the tropical oceans between approximately 15° north and 15° south latitude, the heat absorbed from the sun warms the water. The annual average temperature of the surface layer throughout the region varies from about 27°C to about 29°C.

Research conducted by Mega L. Syamsuddin (2014) on "OTEC Potential in The Indonesian Seas" aims to identify and outline the potential and the position of marine energy source that can be converted into electrical energy in Eastern Indonesia by using and processing of secondary data obtained from satellites (Syamsuddin et al., 2015). From the research get data as in **Table 2.**

Table 2. Carnot efficiency result
(Source: Mega L. Syamsuddin, 2014)

Location	Tw (°C)	Tc (°C)	ΔT	Depth (m)	Carnot Efficiency
South Kalimantan	28.82	7.71	22.11	500	0.73
North Sulawesi	29.22	7.44	21.78	500	0.74
Timur Strait	28.83	6.72	22.11	600	0.76
Makassar Strait	28.83	6.72	22.11	600	0.76
South Sulawesi	28.47	6.18	22.29	700	0.78
West Papua	28.16	6.76	21.4	600	0.75
Morotai Sea	28.47	6.82	21.65	600	0.76

From **Table 2.** can be concluded that potential OTEC in the Indonesian seas is very large, based on some of the areas that have optimum temperature conditions for ocean thermal energy conversion.

2.2 Ocean Thermal Energy Conversion (OTEC)

OTEC or Ocean Thermal Energy Conversion uses the temperature difference between deep and shallow seawater that rotate generators to produce electrical energy. OTEC power systems may be divided into three categories: closed cycle, open cycle, and hybrid cycle. The three cycles are based on a Rankine cycle of heat energy stored in the seawater into electrical energy (Avery & Wu, 1995).

II.2.1. History of Ocean Thermal Energy Conversion (OTEC)

In 1870, the concept of ocean thermal energy conversion (OTEC), was introduced by Jules Verne in Twenty Thousand Leagues Under the Sea.

Within a decade, American, French and Italian scientists are said to have been working on the concept but the Frenchman, physicist Jacques-Arsene d'Arsonval, is generally credited as the father of the concept for using ocean temperature differences to create power. **Table 3**. provides a chronological summary of important advances in OTEC technology.

Table 3. Achievment of OTEC
(Source: Kobayashi et al,)

Years	Achievement
1881	Mr. J. D'Arsonval developed his idea of OTEC theory
1926	G.Claude (France) started experiments of OTEC
1933	Mr. G. Claude generated a net 12 kW output OTEC near Cuba
1964	Anderson (USA) presented a proposal of Off-shore type OTEC
1970	New Energy Research Committee researched OTEC technology (JAPAN
1974	ERDA project started to research OTEC (USA)
1974	First OTEC conference was held (USA)
1977	Saga University succeeded with 1 kW experimental plant
1979	"Mini-OTEC" used cold-water pipe to produce 15kW power (52kW gross)
1980	U.S. DOE built a test site for closed-cycle OTEC heat exchangers, OTEC-1. Results showed that OTEC systems can operate from floating platforms with little effect on the marine environment. The same year 2 laws were enacted to promote OTEC development: Ocean Thermal Energy Conversion Act and Ocean Thermal Energy Conversion Research, Development, and Demonstration Act.
1981	Tokyo Electric Co., and its subsidiary undertook successful experiment of a 120 kW OTEC in Nauru. Used cold-water pipe on the sea bed at 580m depth. Freon was the working fluid. Produced 31.5 kW of net power.
1984	DOE developed a vertical-spout evaporator that converts warm seawater to steam with efficiencies as high as 97%
1985	A 75 kW experimental OTEC plant was installed at Saga University.

1993	USA completed their 210kW open cycle OTEC demonstration facility off coast of Kona, Hawaii.
2009-2013	Lockheed Martin's Alternative Energy Development team has partnered with Makai Ocean Engineering to complete the final design phase of a 10 MW closed cycle OTEC pilot system which will become operational in Hawaii in the 2012-2013 time frame. This system is being designed to expand to 100- MW commercial systems in the near future.

II.2.2. Open Cycle OTEC

In the open-cycle system, the working fluid is vented after use, as shown in **Figure 1**. In this case, the working fluid is water vapor. open cycle OTEC consists of several steps as follows:

1. Flash Evaporator
   At this stage, warm seawater is pumped into a chamber in which the pressure is reduced by a vacuum pump to a value low enough to cause the water to boil. In the flash evaporator, sea water will be separated into two fractions, water (H2O) and salt (CaCO3).
2. Turbine
   The low-pressure water vapor obtained from the flash evaporator will rotate the turbine. The mechanical energy obtained will then be forwarded to the electric generator to produce electricity.
3. Electric Generator
   Electric generators function to convert the mechanical energy obtained from the turbine into electrical energy.
4. Condensor
   The low-pressure steam, after passing through a turbine, is condensed by cold water in a similar chamber and is then discharged into the ocean. Instead of being condensed by direct contact with cold water, the vapor may be directed to a heat exchanger cooled by the cold seawater. In this case, the condensed vapor becomes a source of fresh water.

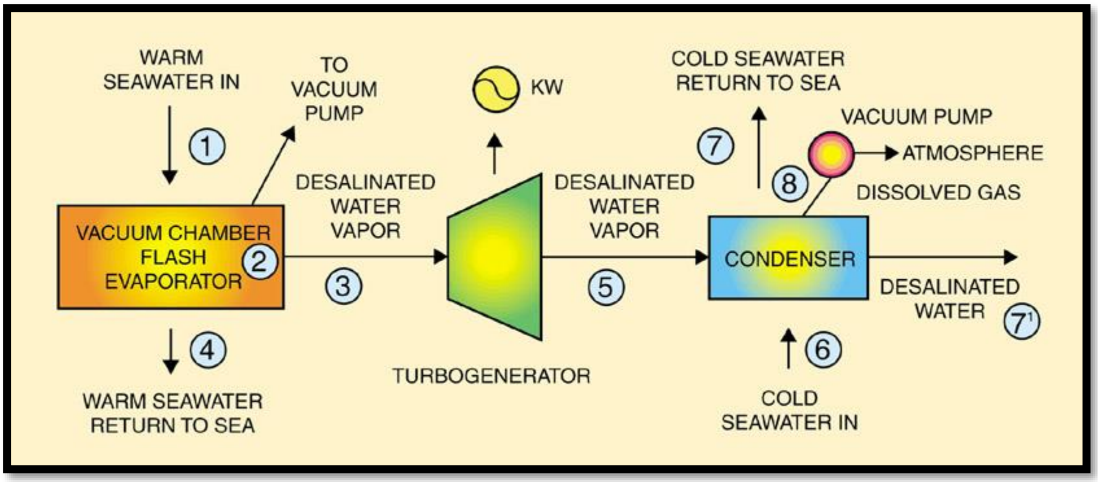


Figure 1. Open Cycle OTEC
(Source: http://www.zoombd24.com/ocean-thermal-energy-conversion-otec-system-working-principles-and-efficiency/)

II.2.3. Close Cycle OTEC

In closed-cycle operation, the working fluid is conserved (i.e., pumped back to the evaporator after condensation), as shown in **Figure 2**. Ammonia is commonly used in this cycle because it has a relatively low boiling point with seawater as a fluid to evaporate and condense. close cycle OTEC consists of several steps as follows:

1. Evaporator
   In the evaporator, seawater warm temperature about 26-30°C will meet with ammonia or other working fluids. There is a heat transfer between the two fluids that causes ammonia to evaporate into high-pressure steam.
2. Turbine
   The high pressure vapor ammonia will then through the turbine and rotate the blades in the turbine.
3. Electric Generator
   Electric generators function to convert the mechanical energy obtained from the turbine into electrical energy.
4. Condensor
   Ammonia vapor that passes through the turbine will get a decrease in temperature and pressure which then continues into the condenser. In the condenser there will be heat transfer between ammonia vapor and cold seawater so that condensation occurs and the ammonia phase changes become saturated liquid.

5. Working Fluid's Pump
   The ammonia passes through condenser will be pumped at a certain pressure toward the evaporator.

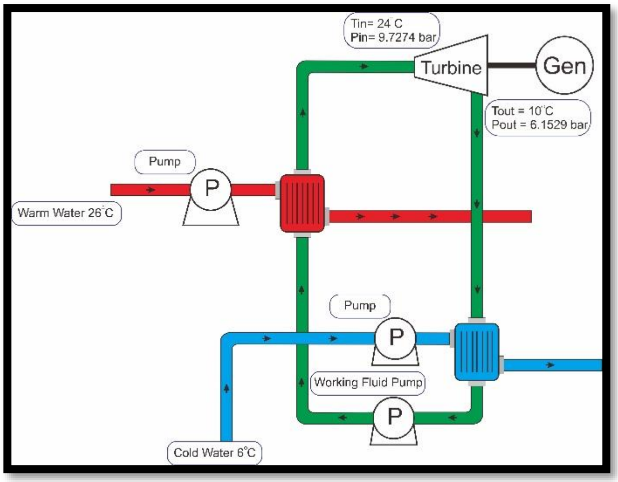


Figure 2. Close Cycle OTEC

II.2.4. Hybrid Cycle OTEC

A hybrid cycle combines the features of the closed- and open-cycle systems. In a hybrid, warm seawater enters a vacuum chamber and is flash-evaporated, similar to the open-cycle evaporation process. The steam vaporizes the ammonia working fluid of a closed-cycle loop on the other side of an ammonia vaporizer. The vaporized fluid then drives a turbine to produce electricity. The steam condenses within the heat exchanger and provides desalinated water.

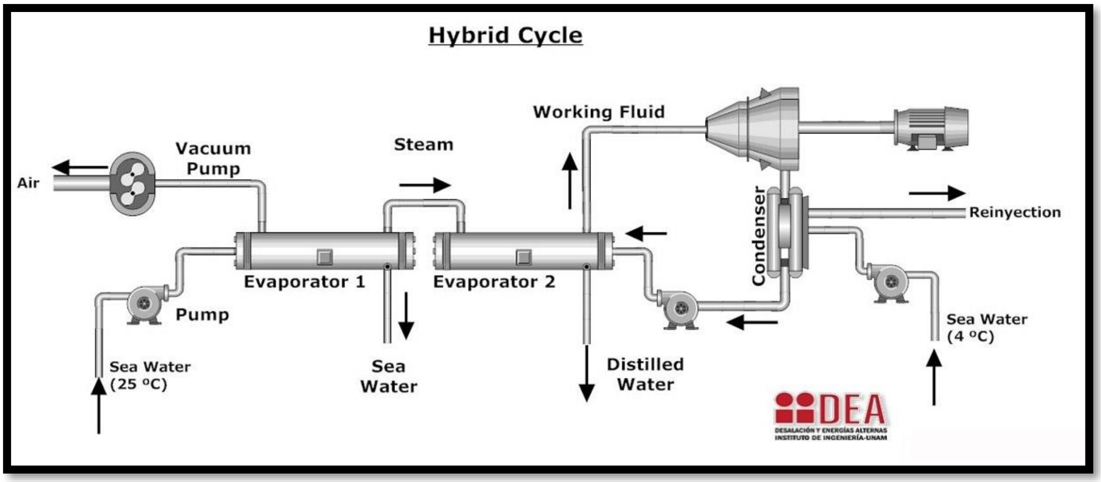


Figure 3. Hybrid Cycle OTEC
(Source: http://proyectos2.iingen.unam.mx/IIDEA/otec_plants.html)

2.3 OTEC Structure

Ocean Thermal Energy Conversion (OTEC) has 2 types of building structures, which are shore-based and off-shore based. The shore-based structure has three main advantages over those located in deep water. Plants constructed on shore do not require sophisticated mooring, lengthy power cables, or the more extensive maintenance associated with open-ocean environments. they can be installed in a sheltered area and relatively save from heavy storm and wave. The shore-based structure also can be integrated with cooling system around plants, desalinated system, and mariculture. Off-shore based structures installed floating in the sea. They have potentially optimal for large systems but has difficultness such as on mooring system, and delivery power system.

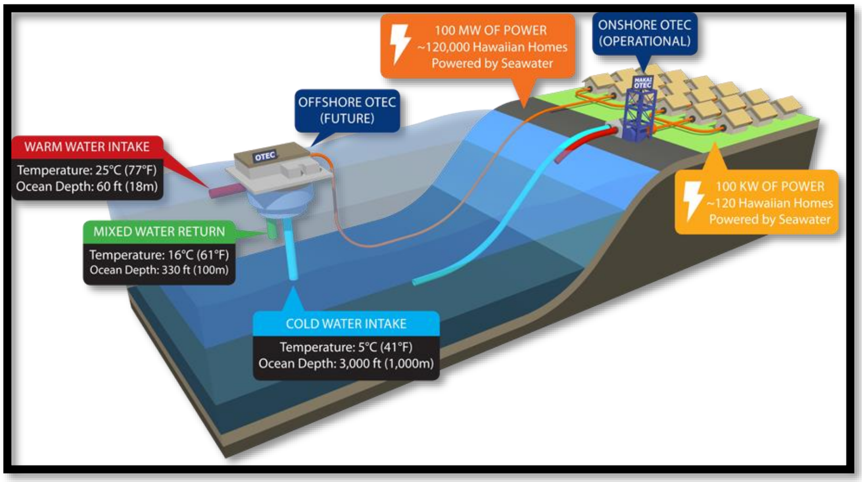


Figure 4. OTEC Shore & Off-Shore Base Structure
(Source: http://www.makai.com/images/Offshore_OTEC_Diagram_900x489.png)

2.4 Rankine Cycle

The Rankine cycle closely describes the process by which steam-operated heat engines commonly found in thermal power plants. This cycle used two phases of working fluid, there are liquid and vapour. in a simple Rankine cycle consists of 4 main components namely condenser, pump, boiler, and turbine as shown in **Figure 5.** The difference of OTEC power plant and thermal power plant is the boiler replaced by evaporator and the boiling point of OTEC cycle is lower so that water is not suitable for working fluid.

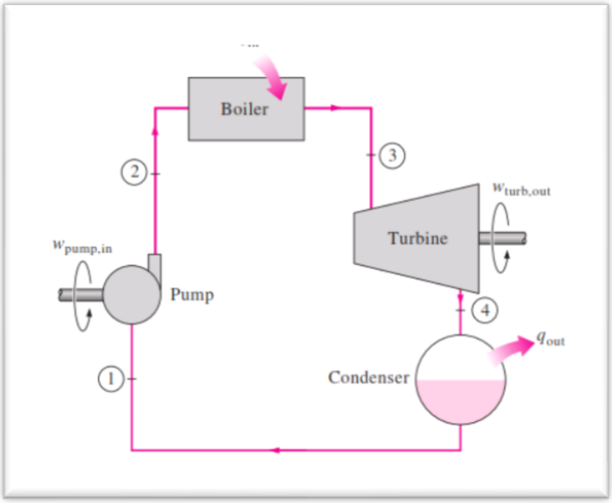
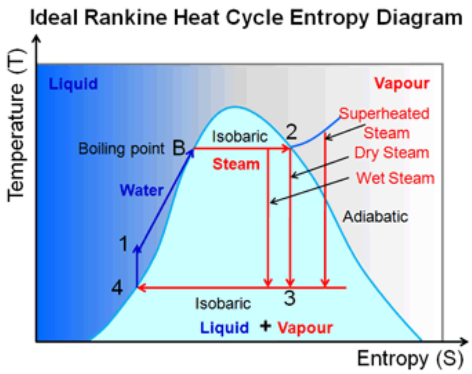


Figure 5. Simple Rankine Plant
(Source: http://sounak4u.weebly.com/vapour--combined-power-cycle.html)



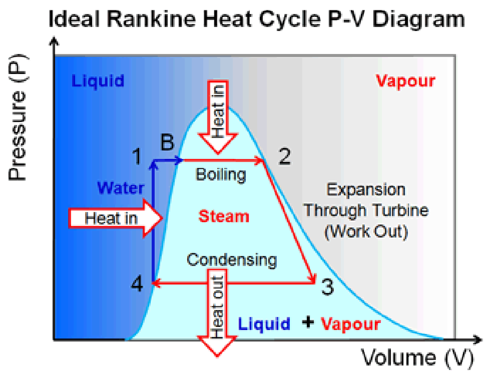


Figure 6. T-S & P-V Diagram of Ideal Rankine Cycle
(Source: https://physics.stackexchange.com/questions/164605/rankine-cycle-pressure)

On ideal Rankine cycle, expansion and compression process occurs isentropic, while the heat transfer process in the condenser or boiler occurs isobaric. There are four processes in the Rankine cycle. These states are identified by numbers in the above T–s and P-v diagram.

- 1-2: Heat transfer process to the working fluid from the boiler with constant pressure (isobaric)
- 2-3: An isentropic expansion occurs on the fluid through the turbine from the superheated state to the condenser pressure.
- 3-4: Heat transfer process of working fluid flowing with constant pressure (isobaric) on condenser to saturated liquid in state 4
- 4-1: The isentropic compression process at the pump where from the saturated liquid state to the compression liquid state.

In the design of the turbine states, if the working fluid conditions inlet (T_1, P_1) and outlet (T_2, P_2) are known, then the inlet enthalpy and entropy can be found by using the properties table. The ideal state ($s_{2s} = s_1$) of the system can be calculated by using the equation (Reynolds & Perkins, 1989):

$$s_{2s} = s_f + x_{2s}(s_g - s_f) \quad \text{(2-1)}$$

$$h_{2s} = h_f + x_{2s}.h_{fg} \quad \text{(2-2)}$$

$$x_{2s} = \frac{s_1 - s_f}{s_g - s_f} \quad \text{(2-3)}$$

Where:

S_{2s} = Entropi ideal state (kj/kg.K)

S_f = Entropi saturated liquid (kj/kg.K)

S_g = Entropi saturated gas (kj/kg.K)

h_{2s} = Entalpi ideal state (kj/kg)

h_f = Entalpi saturated liquid (kj/kg)

h_g = Entalpi saturated gas (kj/kg)

x = Gas quality

The real states can be calculated by using the equation:

$$h_2 = h_1 - \eta_t(h_1 - h_{2s}) \tag{2-4}$$

$$x_2 = \frac{h_2 - h_f}{h_{fg}} \tag{2-5}$$

$$s_2 = s_f + x(s_g - s_f) \tag{2-6}$$

$Where$

S_2	= Entropi real state (kj/kg.K)
S_f	= Entropi saturated liquid (kj/kg.K)
S_g	= Entropi saturated gas (kj/kg.K)
h_2	= Entalpi real state (kj/kg)
h_f	= Entalpi saturated liquid (kj/kg)
h_g	= Entalpi saturated gas (kj/kg)
x	= Gas quality
η_t	= Isentropic efficiency

2.5 Organic Rankine Cycle

Organic Rankine Cycle is a modified energy conversion process from Rankine Cycle. Rankine Cycle usually use pressurized and high-temperature water as a working fluid. While in ORC, the boiling point of this cycle is lower so that water is not suitable for working fluid. Therefore, hydrocarbons or refrigerants are used which have low boiling point as working fluids. The main component of the ORC are condenser, pump, evaporator, and turbine as shown in **Figure 7.**

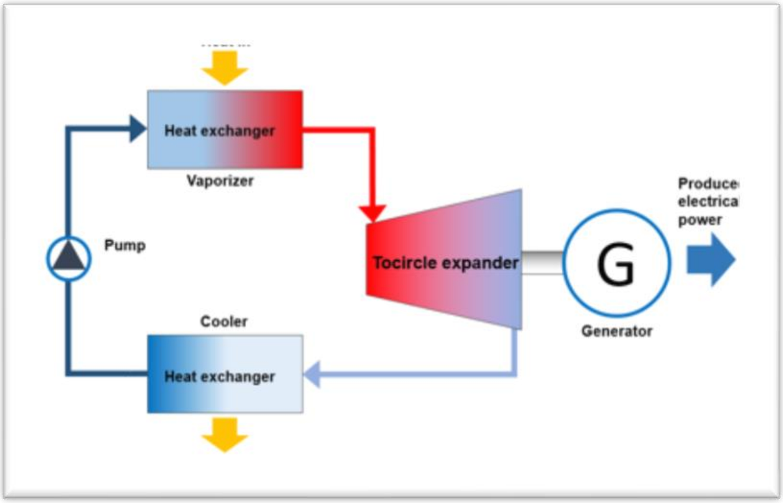


Figure 7. Organic Rankine Cycle Components
(Source: http://www.tocircle.com/applications/energy-efficiency/organic-rankine-cycle/)

2.6 Steam Turbine

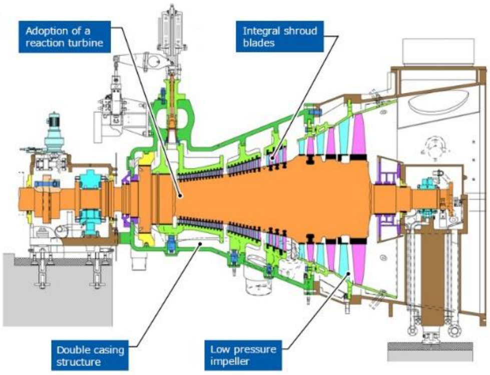


Figure 8. Steam Turbine
(Source: https://www.fujielectric.com/products/thermal_power_generation/)

Figure 8. shown the steam turbine. A turbine is a rotary mechanical device that extracts energy from a fluid flow and converts it into useful work. The rotating part is the rotor, while the non-rotating part is the stator or turbine housing. A turbine is a turbomachine with at least one moving part called a rotor assembly, which is a shaft or drum with blades attached. Moving fluid acts on the blades so that they move and impart rotational energy to the rotor. The work produced by a turbine can be used for rotating the load (i.e. electrical generator, pump, compressor, propeller, etc.). The working fluid may be water, steam, or gas (Arismunandar, 2004).

The closed OTEC cycle is basically the same as the conventional Rankine cycle employed in steam engines, in which the steam is condensed and returned to the boiler after driving a piston or steam turbine. OTEC differs by using a different working fluid and lower pressures and temperatures(Avery & Wu, 1995).

Steam turbine is a driving force that converts steam potential energy into kinetic energy and then converted into mechanical energy in the form of turbine spinning. Turbine axle, directly or with reduction gear, connected with the mechanism to be driven.

2.6.1 History of Steam Turbine

The idea of steam turbine was proposed by Hero of Alexandria, during the 1st century CE. Hero of Alexandria created the first prototypes of turbine called aeolipile as shown in **Figure 8** by the reaction principle. In this device, steam was supplied through a hollow rotating shaft to a hollow rotating sphere. It then emerged through two opposing curved tubes, just as water issues from a rotating lawn sprinkler. The device was little more than a toy, since no useful work was produced.

Another steam-driven machine, proposed by Giovanni Branca in 1629 in Italy, was designed in such a way that a jet of steam impinged on blades extending from a wheel and caused it to rotate by the impulse principle, as shown in **Figure 9**. Patent by James Watt in 1784 starting the developer of the steam engine, a number of reaction and impulse turbines were proposed, all adaptations of similar devices that operated with water.

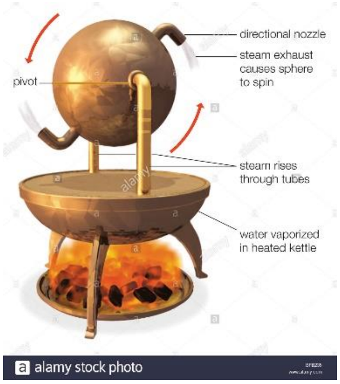


Figure 9. Aeolopile
(Source: http://c8.alamy.com/comp/BFB238/herons-aeolipile-BFB238.jpg)

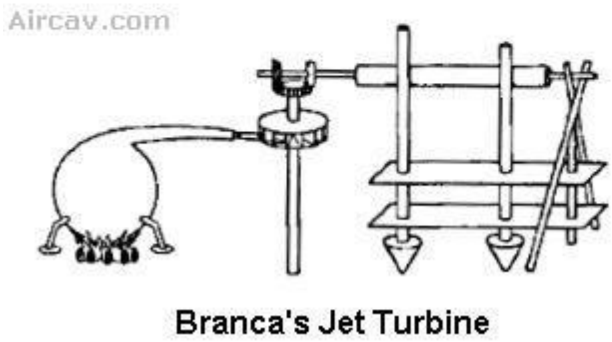


Figure 10. Branca’s Turbine
(Source: http://c8.alamy.com/comp/BFB238/herons-aeolipile-BFB238.jpg)

Small reaction turbines that turned at about 40,000 RPM to drive cream separators constructed by Carl G.P. de Laval of Sweden during 1880s. Their high speed, however, made them unsuitable for other commercial applications. From 1889 to 1897 de Laval built many turbines with capacities from about 15 to hundred horsepower. His 15-horsepower turbines were the first employed for marine propulsion (1892). in 1890s, C.E.A. Rateau of France developed multistage impulse turbine. At about the same time, the velocity-compounded impulse stage was developed by Charles G. Curtis of the United States.

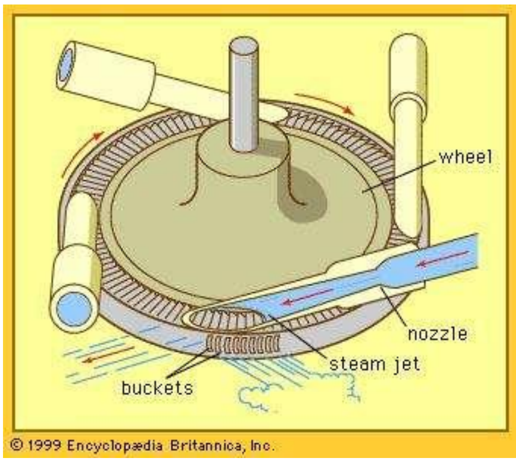


Figure 11. De Laval Turbine
(Source: https://www.britannica.com/technology/turbine/History-of-steam-turbine-technology)

2.6.2 Classification of Steam Turbine

Steam turbines can be classified by:

1. Steam Flow Direction

a. Axial turbine, turbine with steam flow direction parallel to the axis of the shaft (Rajput, 2006).
b. Radial turbine, turbine with steam flow direction perpendicular to the axis of the shaft.

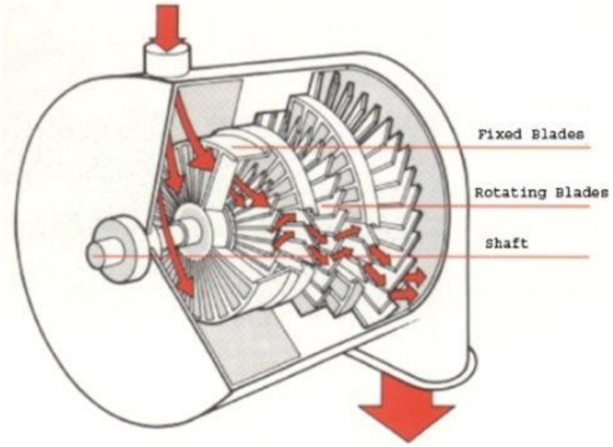


Figure 12. Axial Steam Turbine
(Source: https://www.turbinesinfo.com/wp-content/uploads/2011/07/Schematic-Diagram-of-Parson-Type-Steam-Turbine.jpg)

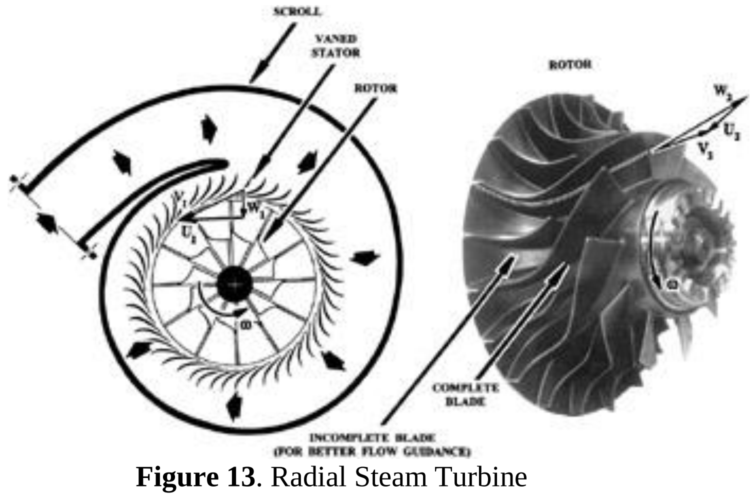


Figure 13. Radial Steam Turbine
(Source: http://images.books24x7.com/bookimages/id_15431/fig418_01.jpg)

2. **The principle used to rotate the turbine wheel through the blade**
   a. Impulse Turbine
   Impulse turbines change the direction of flow of a high-velocity fluid or gas jet. The resulting impulse spins the turbine and leaves the fluid flow with diminished kinetic energy. There is no pressure change of the fluid or gas in the turbine blades (the moving blades), all the pressure drop takes place in the stationary blades (the nozzles). Impulse turbines are most efficient for use in cases where the flow is low and the inlet pressure is high (Rubenstein, Yin, & Frame, 2012).
   b. Reaction Turbine
   Reaction turbines develop torque by reacting to the gas or fluid's pressure or mass. The pressure of the gas or fluid changes as it passes through the turbine rotor blades. Reaction turbines are better suited to higher flow velocities or applications where the fluid head (upstream pressure) is low.

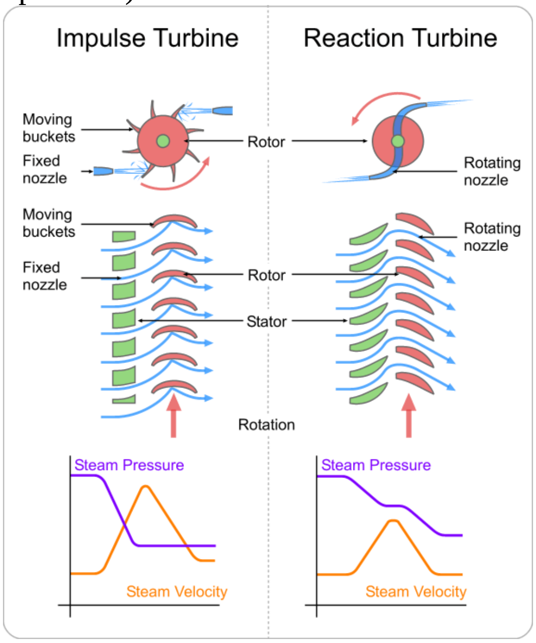


Figure 14. Impulse and Reaction Turbine
(Source: http://www.mech4study.com/2015/12/difference-between-impulse-and-reaction-turbine.html)

3. **Steam Pressure**
   a. Low pressure turbines, using steam at a pressure of 1.2 to 2 ata.
   b. Medium pressure turbines, using steam at pressure up to 40 ata.
   c. High pressure turbines, utilising steam at pressure above 40 ata.

d. Turbines of very high pressure, utilising steam at pressure of 170 ata and higher.
e. Turbines of supercritical pressure, using steam at pressure above 225 ata.

2.6.3 Steps how to Design a Turbine

To design a single stage turbine, it takes several steps (Shlyakhin, 1999), including:

1. Determining the capacity/power
2. Determining rotation per minutes
3. Determining temperature and pressure inlet
4. Determining pressure outlet

2.6.4 Power Generated by Turbine

Steam turbine is designed by a certain power. The power generated by the turbine is obtained from the difference in enthalpy and the steam capacity entering the turbine, and losses inside the turbine due to energy transformation. The power generated by the turbine can be determined by using the formula (Dietzel & Sriyono, 1996):

$$P = h . \dot{m}_s . \eta_i . \eta_m \quad \text{.......................................(2-7)}$$

Where:

P	= Power (kW)
h	= Difference of entalpi (KJ/Kg)
$\dot{m}_s$	= Steam Capacity (Kg/s)
η_i	= Internal efficiency of the turbine
η_m	= Mechanical efficiency of the turbine (0,94-0,97)

If the torque and RPM of turbine are known, then turbine power can be calculated using the equation:

$$P = T \times 2\pi \times RPM/60000 \quad \text{......................(2-8)}$$

Where:

P	= Power (kW)
T	= Torque (Nm)
RPM	= Revolution per minutes

2.6.5 Velocity Triangle of Turbine

In the steam turbine, vapor is expanded in the nozzle so, obtained the vapor velocity (c_1) that will enter to the rotor on turbine. The rotor rotates with the velocity *u*. It needs c_1 and *u* ratio with a certain value so that the steam flow out of the nozzle works optimally. Thus, can be obtained inlet and outlet angle (Dietzel & Sriyono, 1996). Velocity triangle can be seen in **Figure 14.**

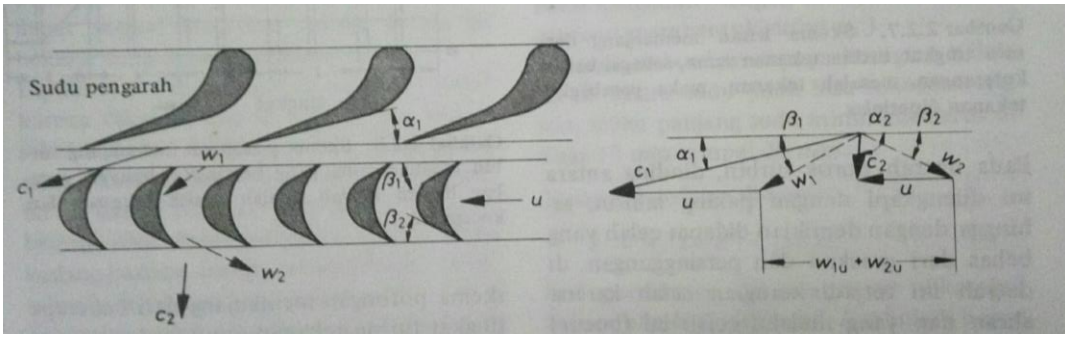


Figure 15. Velocity Triangle
(Source: Dietzel, 1996)

Angle α1 and β1 shall be made in such a way, according to the vapor velocity. Value of α1 is free to determined, but should be as small as possible. The optimum value of α1 is between 14°-20° (Shlyakhin, 1999). From α1 can be found:

$$w_1 = \sqrt{{C_1}^2 + u^2 - 2. C_1. u. \cos \alpha_1} \text{ ....(2-9)}$$

$$\beta_1 = \frac{C_1}{w_1} \times \alpha_1 \text{ ....(2-10)}$$

$$\beta_2 = \beta_1 - (3° - 5°) \text{ ....(2-11)}$$

$$w_2 = \Psi \times w_1 \text{...(2-12)}$$

$$C_2 = \sqrt{{w_2}^2 + u^2 - 2. w_2. u. \cos \beta_2} \text{........................(2-13)}$$

$$Sin\ \alpha_2 = \frac{w_2}{C_2} \text{..(2-14)}$$

Where:

c_1 and c_2 : absolute velocity of steam inlet and outlet from the nozzle
w_1 and w_2 : relative velocity of steam inlet and outlet from the rotating blades
u : circumference velocity of the rotating blade
α_1 and α_2 : angle of the nozzle
β_1 and β_2 : angle of the rotating blades
Ψ : speed coefficient

2.7 Heat Exchanger

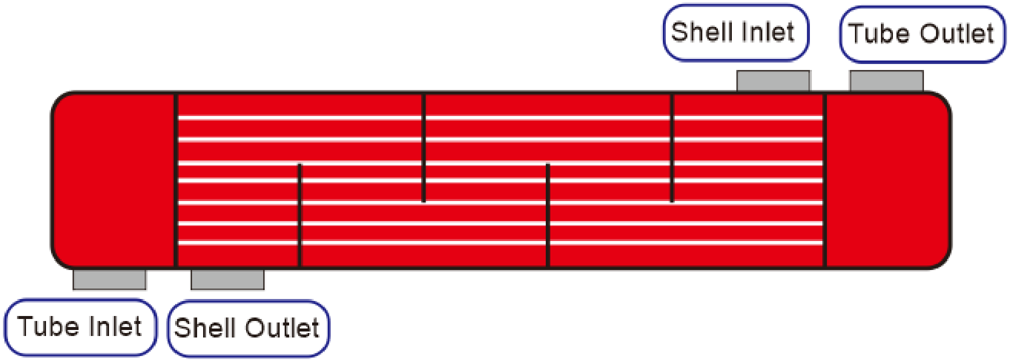


Figure 16. Heat Echanger

Heat exchanger used to transfer heat between two or more fluids. The fluids separated by a solid wall to prevent mixing or they may be in direct contact. They are usually used in space heating, refrigeration, air conditioning, power stations, chemical plants, petrochemical plants, petroleum refineries, natural-gas processing, and other. In OTEC system used heat exchanger that is evaporator and condenser. evaporator replace the boiler function in the steam power plant. An evaporator used to turn the liquid form of a chemical substance such as water into its gaseous-form/vapor. A condenser used to condense a substance from its gaseous to its liquid state, by cooling it. The design calculation for heat exchange means is essentially determining the heat transfer coefficient and heat transfer area (A) of the following equations (Holman, 2010):

$$A = \frac{Q}{U \times LMTD \times F} \quad \text{....................(2-15)}$$

$$Q = \dot{m}_{Warm} \times C_{pWarm} \times \Delta T \quad \text{....................(2-16)}$$

$$LMTD = \frac{(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})}{\ln\left[\frac{(T_{h,in} - T_{c,out})}{(T_{h,out} - T_{c,in})}\right]} \quad \text{....................(2-17)}$$

Where:

A	= Area (m2)
Q	= Heat transfer rate (Watt)
U	= The overall heat transfer coefficient (W/m^2.^{o}C)
LMTD	= Logarithmic mean temperature difference (^{o}C)
F	= LMTD correction factor

2.8 Pump

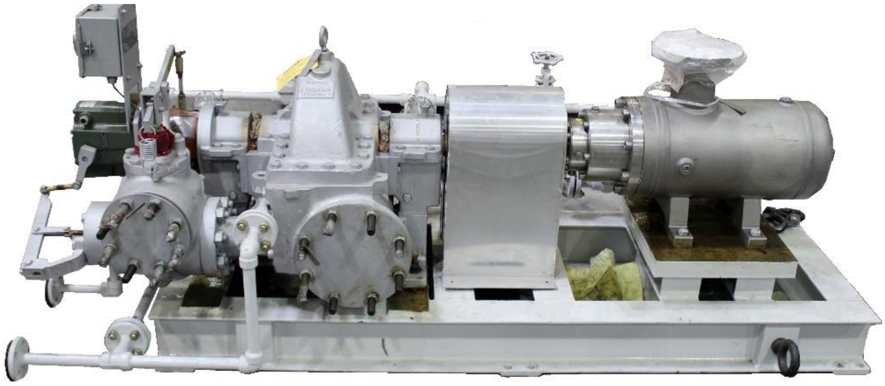


Figure 17. Pump
(Source: http://nuovoparts.com/steam-turbine-driven-allweiler-pump-package/)

A pump is a device that moves fluids by mechanical action. Pumps can be classified into three major groups according to the method they use to move the fluid: direct lift, displacement, and gravity pumps. Pumps operate by some mechanism, usually reciprocating or rotary, and consume energy to perform mechanical work for moving the fluid. Pumps operate via many energy sources, including manual operation, electricity, engines, or wind power, come in many sizes, from microscopic for use in medical applications to large industrial pumps. The characteristics of the pump are determined by the following values:

1. The volume of the pumped fluid (V),
2. Head losses (H),
3. Condition on each side of suction.

2.7.1 Pump Head

One of the important factors in sizing a pump is total head requirements. a pump head is maximum height that the pump can achieve pumping against gravity. Total head is the sum of Head static, Head Pressure, Head Velocity, Head Loss.

1. Head Static
   Head static is the maximum height reached by the pipe after the pump. Head static pump is calculated from the pump inlet till the end of discharge.

2. Head Pressure
   Head Pressure is the diference of pressure on the suction and discharge.

3. Head Velocity
   Head Velocity is difference velocity of fluid between in suction and discharge of pump.

4. Head Loss
   Head loss is energy loss per unit weight of the fluid in the drainage of fluid in the piping system. Head Losses including head major and head minor in suction and discharge.
   - Head Major
     Major losses are associated with frictional energy loss per length of pipe depends on the flow velocity, pipe length, pipe diameter, and a friction factor based on the roughness of the pipe, and whether the flow is laminar or turbulent. Head Major can be calculated with the equation:

$$Major\ losses = \frac{f \times L \times v^2}{d \times 2g} \text{ ....(2-18)}$$

Where:

f	= the Darcy friction factor (unitless)
L	= the pipe length (m)
d	= the hydraulic diameter of the pipe D (m)
g	= the gravitational constant (m/s2)
v	= the mean flow velocity V (m/s)

The value of the Darcy friction factor is determined from the flow type (ie laminar or turbulent), with the equation:

$$f = 0.02 + 0.0005/d \text{ (Turbulent Re>2300)................(2-19)}$$

$$f = \frac{64}{Re} \text{ (Laminar Re<2300)..(2-20)}$$

$$Re = \frac{v \times d}{\mu} \text{ ....(2-21)}$$

Where:
Re = Reynold Number
v = flow velocity (m/s)
d = inner diameter of pipe (m)
μ = kinematis viscosity (m^2/s)

- Head Minor
  Minor loss is a pressure loss in components like valves, bends, tees and similar. Head minor can be calculated with the equation:

$$Minor\ Losses = \frac{\Sigma nk \times v^2}{2g} \text{..............................(2-22)}$$

Where:
Σnk = Minor loss coefficient

2.7.2 Pump Power

As described in the previous section, the total head has an effect on the size of the pump and the driving machine. The pump power is the power that can be used to move the fluid and drive the driving machine. Pump power can be calculated by using equation:

$$P = \rho \times Q \times g \times H_{total} \text{...............................(2-23)}$$

Where:
P = Power (Watt)
ρ = Density (kg/m3)
Q = Volume flow rate (m3/s)
g = the gravitational constant (m/s2)
H_{total} = Total head (m)

2.9 Computational Fluid Dynamics (CFD)

Computational fluid dynamics (CFD) is a branch of fluid mechanics that uses numerical analysis and data structures to solve and analyze problems that involve fluid flows. Computers are used to perform the calculations required to simulate the interaction of liquids and gases with surfaces defined by boundary conditions (CFD-Online, n.d.).

In the simulation process, there are three steps that must be done, there are pre-processing, solving and post-processing.

1. Pre-Processor
   Pre-processor is the initial stage in Computational Fluid Dynamic (CFD) which is the stage of data input that includes the determination of domain and boundary condition. At this stage, meshing is also done, where the analyzed object is divided into the number of specific grids.
2. Processor
   The next step is the processor stage. At this stage, is done the process of calculating data that has been entered using iterative related equations until the results obtained can reach the smallest error value.
3. Post Processor
   The last step is the post processor stage, the results of the calculations at the processor stage will be displayed in pictures, graphs and animations.

CHAPTER III
METHODOLOGY

Methodology represents the completion steps in this study. Problem solving in this research used Computational Fluid Dynamics method. The steps of this methodology are as follows as in **Figure 18.**

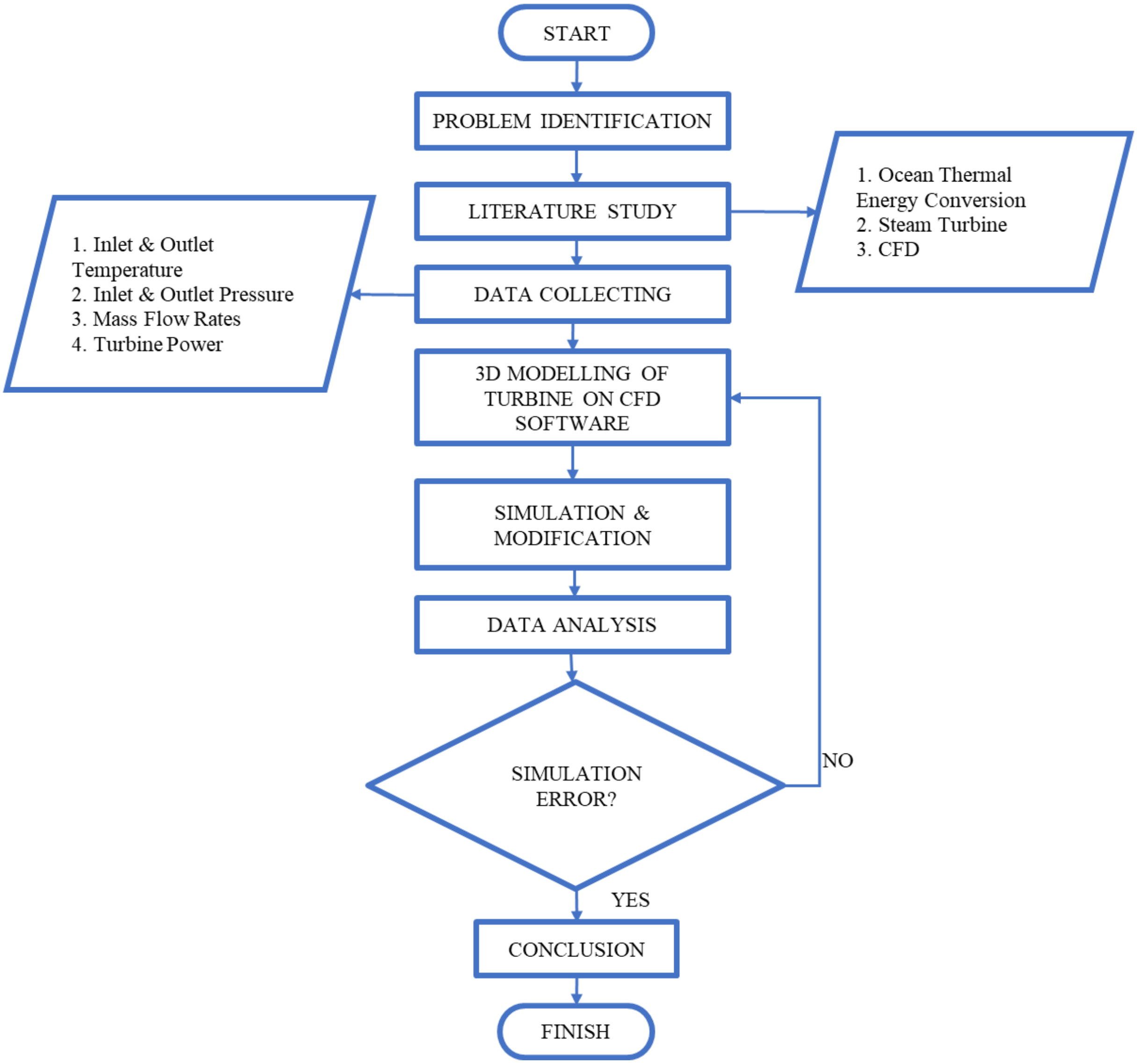


Figure 18. Methodology Chart

3.1 Problem Identification

The first step in this final project is identifying the problem. The selection of the topic starts from Indonesia faces a decreasing in fossil energy reserves and Indonesia's dependence on fossil energy which is not been matched by the discovery of new energy reserves. Therefore, it is necessary to increase the use of renewable Energy to meet energy needs. One of many renewable energies is OTEC. The important component of the Ocean Thermal Energy Conversion (OTEC) system is the turbine, therefore further research is needed on OTEC turbine optimization.

3.2 Literature Study

Literature study aims to get a summary of the basics of existing theories, references and various information that can be a supporter in this final project. Literature study was conducted by collecting references to be studied as supporting materials for research activities such as books, journals, papers or from internet. In addition, it can also discussing with competent persons in this field. The literature studied is closely related to: Ocean Thermal Energy Conversion (OTEC), Steam Turbine, Heat Exchanger, Pump, and Computational Fluid Dynamics (CFD).

3.3 Data Collecting and Planning

At this stage will be data collection and planning for Ocean Thermal Energy Conversion (OTEC) turbine, the required data are shallow and deep water temperature in order to know the potential of OTEC in Indonesia that will be used as reference for turbine planning. Data obtained from research conducted by Mega L. Syamsuddin (2014) on "OTEC Potential in The Indonesian Seas" the average temperature of shallow and deep water temperature are 28.68°C and 6.9°C with the difference reaching 21.78°C at a depth 500 – 700m. From the data assumed the inlet and outlet of temperature and pressure of turbine are:

Tin	: 24°C
Pin	: 9.7274 bar
Tout	: 10°C
Pout	: 6.1529 bar

3.4 Design of Turbine in CFD Software

After the blade’s angle calculation, then the next step is to design turbine into 3D form by using the software. In this research, 13 variations of turbine model will be made, that are: Single Stage, 2 Stage, 2 Stage 5 Degree, 2 Stage 10 Degree, 2 Stage 15 Degree, 2 Stage 20 Degree, 2 Stage 25 Degree, 2 Stage 30 Degree, 2 Stage 35 Degree, 2 Stage 40 Degree, 2 Stage 45 Degree, 2 Stage 50 Degree, 3

Stage 40 Degree. **Figure 19**. Is a side view of the turbine model. The number of turbine blades can be seen on **Table 4.**

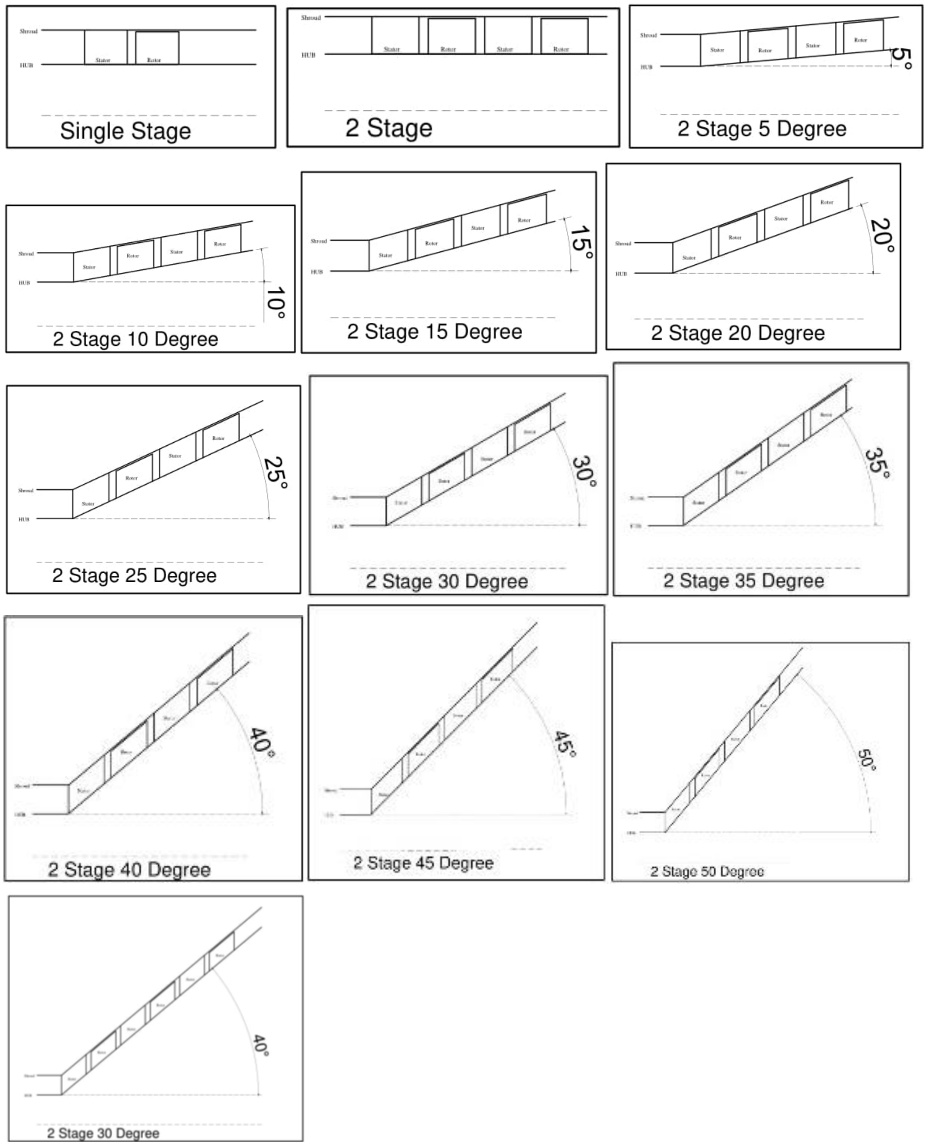


Figure 19. Side View of Turbine models

Table 4. Number of Blades

Name	Row	Number of Blades
Single Stage	1	10
	2	15
2 Stage	1	10
	2	15
	3	15
	4	20
3 Stage	1	10
	2	15
	3	15
	4	20
	5	20
	6	25

In this study, the software used to design the turbine is Autoblade.Design of turbine as shown in **Figure 20.** single blade view and **Figure 21.** Full blade view for single stage turbine. The other OTEC turbine model, it can be seen in Attachment 2.

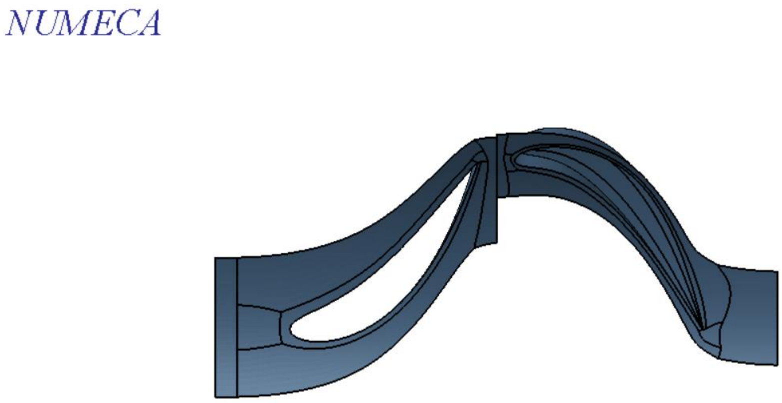


Figure 20. Design of turbine single blade view

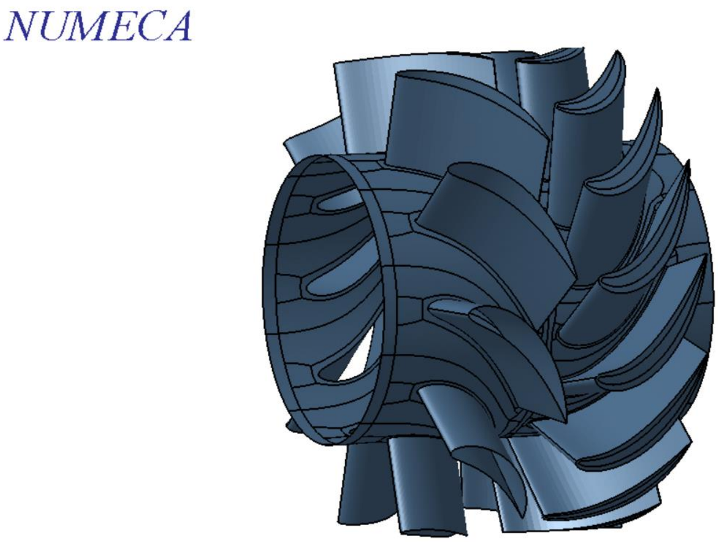


Figure 21. Design of turbine full view

3.5 Simulation

Simulation and modification of model is done by using Computational Fluid Dynamics (CFD) software. In this research, the turbine model is simulated and will be compared to get the highest efficiency and net power.

3.6 Data Analysis

The data has been obtained from the simulation process, then do the calculation of heat exchanger, pump power, and net power, analyzed and compared the efficiency.

3.7 Conclusion

The last step in this final project is making conclusion. The conclusion based on the results of simulation and analysis. This conclusion is prepared to answer the research problem.

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CHAPTER IV
DATA ANALYSIS

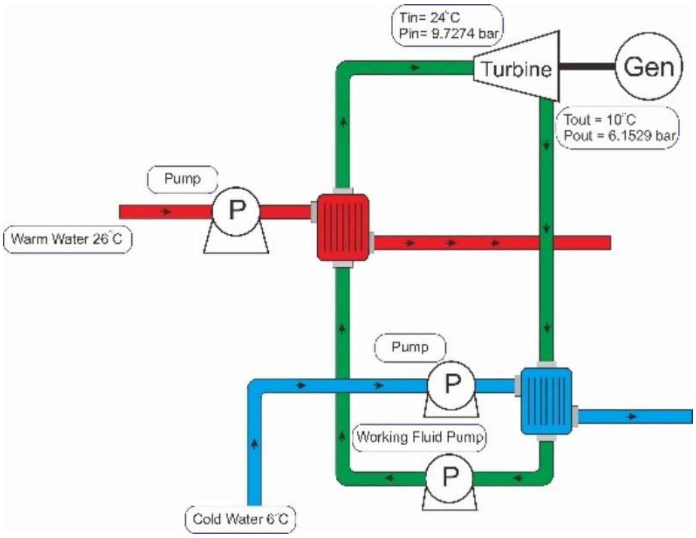

Figure 22. Close cycle OTEC System

OTEC system consists of turbines, generators, evaporators, condensers, and pumps as shown on **Figure 22.** In this chapter will explain how to design Ocean Thermal Energy Conversion (OTEC) turbine, simulation of turbine design, heat exchanger calculation, pump calculation, and net power calculation to be delivered to the consumer.

4.1 Design and Drawing of OTEC's Turbine

In this sub-chapter will be explaining the design process using Autoblade software and simulation process using NUMECA Fine Turbo software. The first process is to determine the working fluid state at the inlet and outlet of OTEC's turbine, then, determining the angle of the turbine blades, and the last is the drawing of OTEC's turbines by using Autoblade software.

4.1.1 Calculation of Working Fluid State

Data obtained from research conducted by Mega L. Syamsuddin (2014) on "OTEC Potential in The Indonesian Seas" the average temperature of shallow and deep water temperature are 28.68°C and 6.9°C with the difference reaching 21.78°C at a depth 500 – 700m. From the data assumed the inlet and outlet of temperature and pressure of turbine are:

Tin : 24°C
Pin : 9.7274 bar
Tout : 10°C
Pout : 6.1529 bar

Then, State 1 is found after the working fluid exits the evaporator. From the ammonia table properties **Attachment 1** we get:

P_1 : 9.7274 bar
T_1 : 24°C
h_1 : 1462.61 kJ/kg
s_1 : 5.0394 kJ/kg.K

State 2 is found after the working fluid flow through the turbine can be determined by equation 2.1-2.7 :

P_2 : 6.1529 bar
T_2 : 10°C

State $s_{2s} = s_1$

- $s_{2s} = s_f + x(s_g - s_f) \rightarrow x_{2s} = \frac{s_1 - s_f}{s_g - s_f}$

$$x_{2s} = \frac{5.0394 - 0.8769}{5.2033 - 0.8769} = 0.962116$$

$$s_{2s} = 0.8769 + 0.962116(5.2033 - 0.8769)$$
$$= 5.0394 \text{ kJ/kg.K}$$

- $h_{2s} = h_f + x.h_{fg}$

$$h_{2s} = 226.75 + 0.962116 \times 1225.03 = 1402.371 \text{kJ/kg}$$

- $\eta_t = \frac{h_1 - h_2}{h_1 - h_{2s}} = 50\%$ (assumption)

Isentropic efficiency for turbine as low as 40% (Agency, 2015)

- $h_2 = h_1 - \eta_t(h_1 - h_{2s})$

$$h_2 = 1462.61 - 50\% \times (1462.61 - 1405.371)$$
$$h_2 = 1433.991 \text{ kJ/kg}$$

- $x_2 = \frac{h_2 - h_f}{h_{fg}}$

$$x_2 = \frac{1433.997 - 226.75}{1225.03} = 0.9855$$

- $s_2 = s_f + x\left(s_g - s_f\right)$

$$s_2 = 0.8769 + 0.9855(5.2033 - 0.8769) = 5.1404 \text{ kJ/kg.K}$$

From the calculation above, can be found:
State 1 is found after the working fluid exits the evaporator:

P_1 : 9.7274 bar
T_1 : 24°C
h_1 : 1462.61 kJ/kg
s_1 : 5.0394 kJ/kg.K

State 2 is found after the working fluid flow through the turbine can be determined:

P_2 : 6.1529 bar
T_2 : 10°C
h_2 : 1433.997 kJ/kg
s_2 : 5.1404 kJ/kg.K

4.1.2 Calculation of Blade's Angle

The angle of the turbine blades is designed so as to produce optimum work on the turbine. Value of α1 is free to determined, but should be as small as possible. The optimum value of α1 is between 14°-20° (Shlyakhin, 1999). In this research, the value of α1 = 15 °. Following is the calculation of blade's angle:

- $\alpha_1 = 15°$
- $C_{1t} = 44.72 \times \sqrt{\Delta h}$

  $C_{1t} = 44.72 \times \sqrt{1462.61 - 1433.991} = 239.24$ m/s
- $C_1 = \varphi . C_{1t}$

  $C_1 = 0.95 \times 239.24 = 227.28$ m/s

  Range of φ is between 0.91-0.98 (usually use 0.95)
- $u = (\frac{u}{c_1}) \times c_1$

  $u = (0.4) \times 227.28 = 90.91\ m/s$

  Value of $(\frac{u}{c_1})$ for single stage turbine is between 0.2-0.6, optimal value is 0.4 (Shlyakhin, 1999).
- $w_1 = \sqrt{{C_1}^2 + u^2 - 2. C_1 . u . \cos \alpha_1}$

  $w_1 = \sqrt{227.28^2 + 90.91^2 - 2 \times 227.28 \times 90.91 \times \cos 15°}$

  $w_1 = 141.4\ m/s$

- $\beta_1 = \frac{C_1}{w_1} \times \alpha$
  $$\beta_1 = \frac{227.28}{141.4} \times 15° = 24.1°$$
- $\beta_2 = \beta_1 - (3° - 5°)$
  $\beta_2 = \beta_1 - (3°) = 21.1°$
- $w_2 = \Psi \times w_1$ ($\Psi = 0.45$) (Dietzel & Sriyono, 1996) page 94)
  $w_2 = 0.45 \times 141.4$
  $w_2 = 63.65\ m/s$
- $C_2 = \sqrt{{w_2}^2 + u^2 - 2.w_2.u.\cos\beta_2}$
  $C_2 = \sqrt{63.65^2 + 90.91^2 - 2 \times 63.65 \times 90.91 \times \cos 21.1}$
  $C_2 = 38.98\ m/s$
- $\sin \alpha_2 = \frac{w_2}{C_2} \times \sin \beta_2$
- $sin\ \alpha_2 = \frac{63.65}{38.98} \times sin\ 24.1°$
  $\alpha_2 = 31.78°$

From the calculation above, we can get the value:

$\alpha_1 =$ 15°
$\alpha_2 =$ 31.78°
$\beta_1 =$ 24.1°
$\beta_2 =$ 21.1°

4.1.3 Drawing Process of OTEC's Turbine

Model drawing process is done to ease the simulation process, it uses Autoblade software. The components drawn are stator and rotor by using the Autoblade software, the components are drawn one by one. In the process of model depiction, performed in several steps: Hub and shroud endwalls definition, stream surface definition, main blade and construction plane.

1. Hub and Shroud Endwalls Definition

   Hub and shroud endwalls are assumed to be axisymmetric surfaces. Corresponding surfaces defined by 2 curves (Z and R), where Z and R are the coordinate point of axial and radial directions. Both hub and shroud can be defined using different method. **Figure 23.** shown the hub and shroud curve type of turbine design. The value of Z and R shown in **Figure 24.**

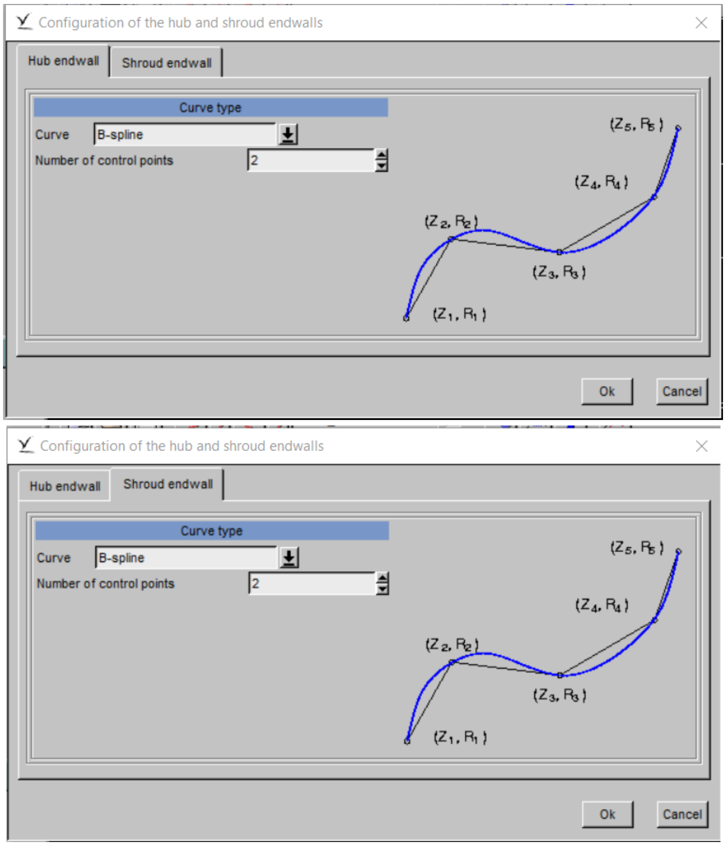

Figure 23. Hub and Shroud Definition

Name	Lower bound	Expression	Value	Upper bound	Reference
HUB_Z1	-0.1		-0.1	0	0.001
HUB_R1	0.05		0.06	0.245	0.001
HUB_Z2	0.1		0.2	0.2	0.001
HUB_R2	0.05		0.06	0.245	0.001
SHROUD_Z1	-0.1		-0.1	0	0.001
SHROUD_R1	0.05		0.1	0.3	0.001
SHROUD_Z2	0.1		0.2	0.2	0.001
SHROUD_R2	0.05		0.1	0.3	0.001

Figure 24. Hub and Shroud value

2. Stream Surface Definition

   Second step menu defines the stream surface type, the rulings and the spanwise location of their traces in the meridional plane. The primary sections are defined on these stream surfaces, so, it defined before the blade sections. In axial turbine use cylindrical for surface set up. The radius of the hub and shroud are determined 0.06 m and 0.1 m. It is intended that when the model is made, it becomes low cost for research. **Figure 25.** shown the configuration of stream surface. **Figure 26.** Meridional view shown the projection of a turbomachine onto the axial-radial (Z,R) plane using a cylindrical frame of reference.

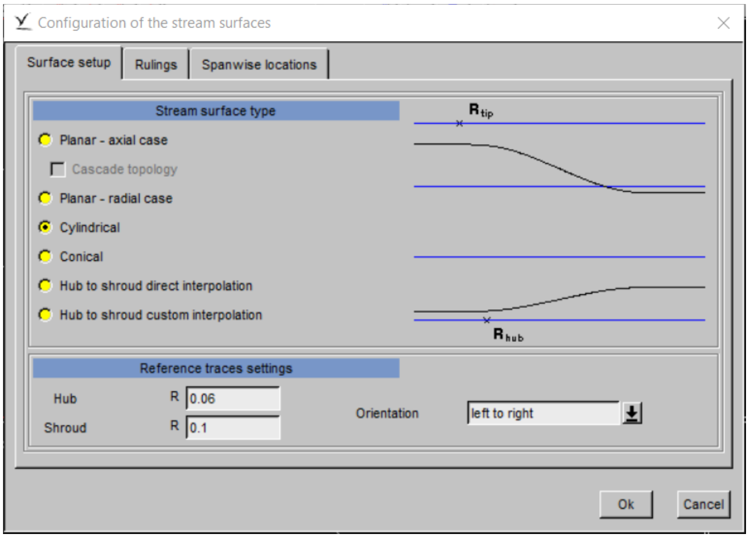

Figure 25. Stream Surface Setup

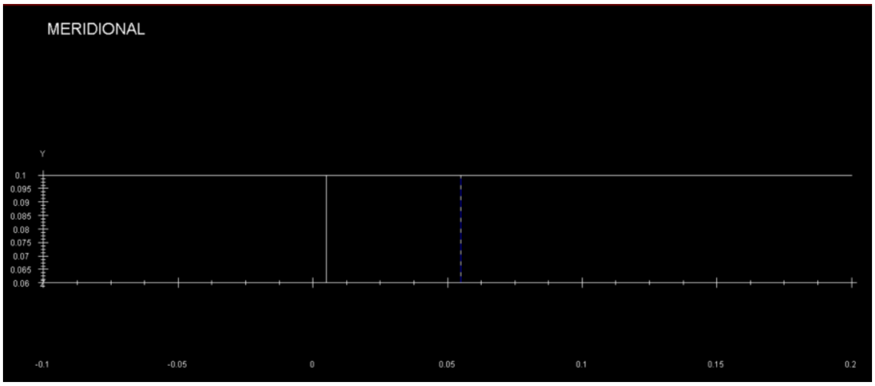

Figure 26. Meridional view

3. Main Blade Construction

Once stream surface setup, the primary blade sections can be constructed and placed on the stream surfaces. In AutoBlade™, any blade section is built from a camber curve in a prescribed 2D construction plane. **Figure 27.** and **Figure 28.** shown the parameter of stator and rotor from the calculation. The projection of the parameter value can be shown on

Camber View in **Figure 29.** and **Figure 30.** While, both **Figure 31.** and **Figure 32.** are the design of OTEC's turbine in 3D view.

Name	Lower bound		Expression	Value	Upper bound	Reference
S1_CAMBER_GAMMA	37.9028618344			45	45	1
S1_CAMBER_BETA1	-2.5563008962			0	3.44369910377	1
S1_CAMBER_BETA2	56.391938744(			75	89	1
S1_MAX_NACA_THICK	0.12			0.145	0.145	0.01

Figure 27. Parameter for stator

Name	Lower bound		Expression	Value	Upper bound	Reference
S1_CAMBER_GAMMA	-89		gamma	-45	4.79985742775	1
S1_CAMBER_BETA1	-89		Inlet Angle	25	89	5
S1_CAMBER_BETA2	-89		Outlet Angle	-68	89	5
S1_MAX_NACA_THICK	0.12		Maximum Thickness	0.145	0.145	0.01

Figure 28. Parameter for rotor

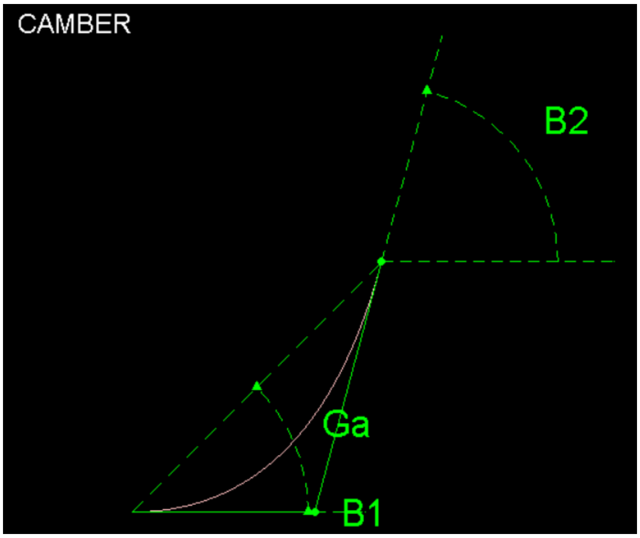

Figure 29. Camber view of stator

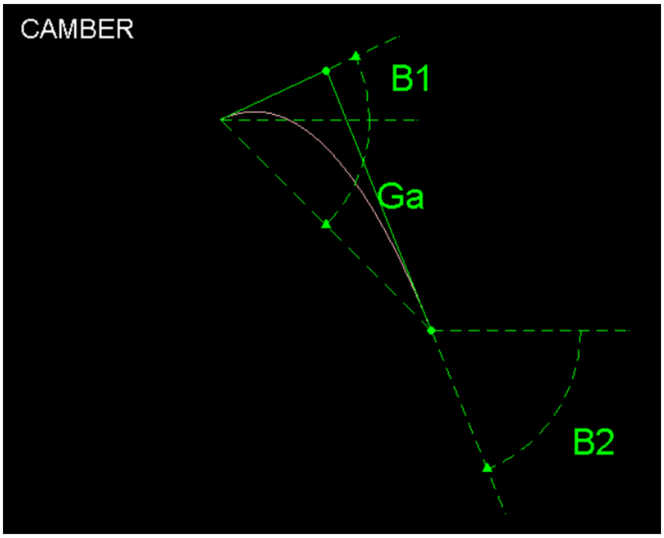

Figure 30. Camber view of rotor

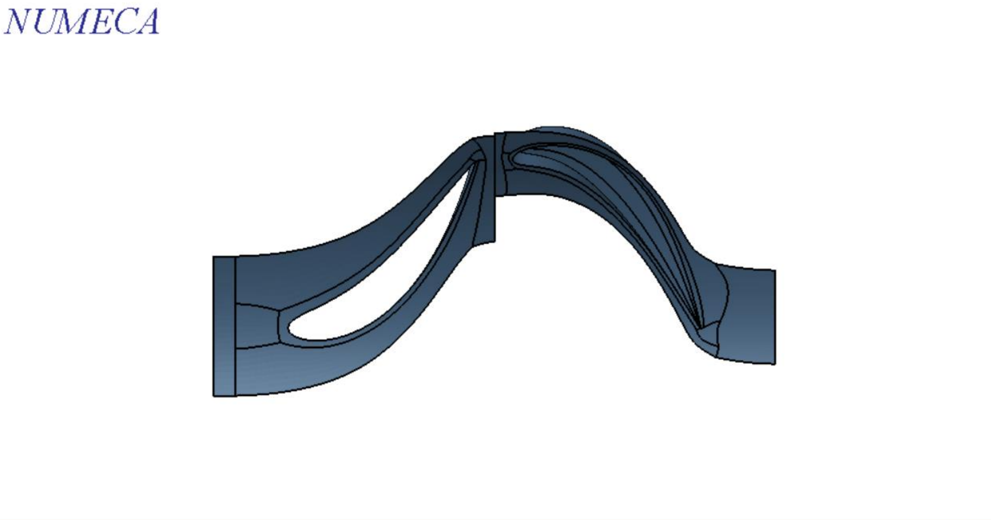

Figure 31. Single blade view

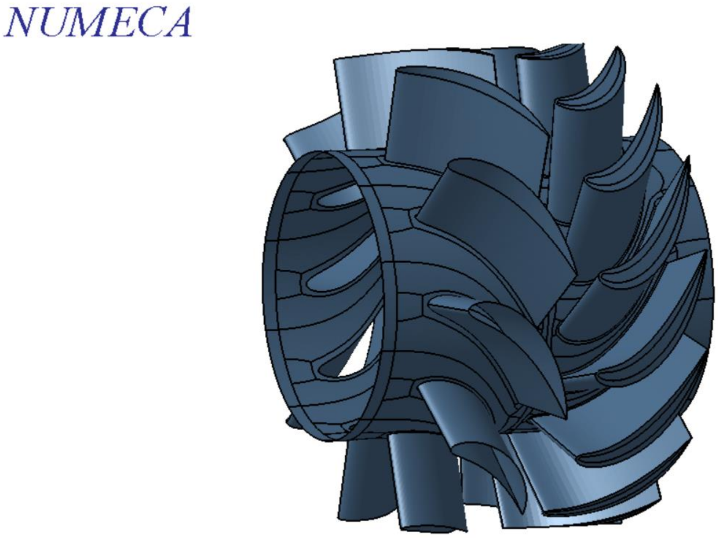

Figure 32. Full blade view

After the drawing process of model to be tested is completed, the next step is simulation and data collecting done with CFD approach using NUMECA FINE Turbo software. However, before the simulation process, mesh generation need to be done by using NUMECA Autogrid software.

4.1.4 Mesh Generation

The meshing process is done to ease the pre-processing process for numerical computation. Pre-processing consists of defining the geometrical description of the to-be-studied model and the discretization (mesh generation) of the to-be-studied domain. In this process using NUMECA AutoGrid5™ . AutoGrid5™ has a user interface that includes multiple windows showing the model visualization as shown in **Figure 33**.

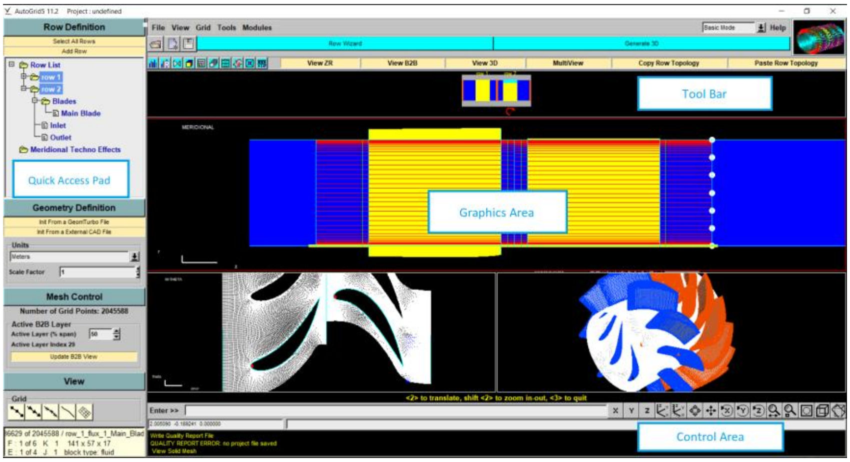

Figure 33. AutoGrid User Interface

The first process is to import stator and rotor files that have been created by using AutoGrid5^{TM} with ".geomturbo" extension and then define hub, shroud, blade, leading edge, and trailing edge. When the definition provide is correct as shown in the **Figure 34**, then, determination of blade row type, number of grid, and rotational speed.

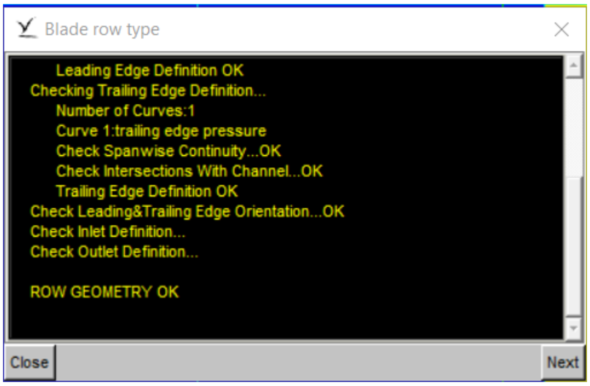

Figure 34. Geometry Check

In this research, row type is axial turbine, number of blades is 10 and 15 for stator and rotor, and rotation speed 3000 rpm for single stage turbine, as in **Figure 35**. Result on mesh generation can be seen on **Figure 36.**

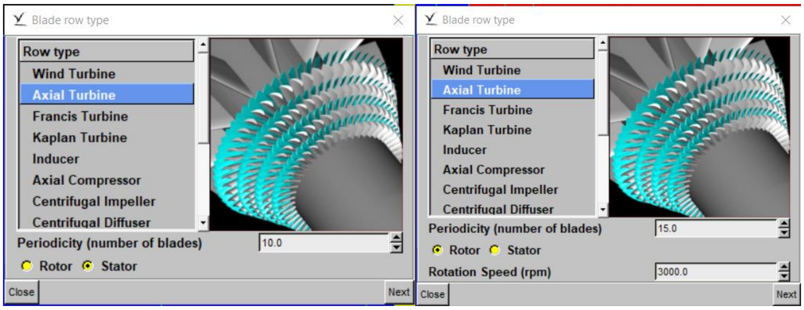

Figure 35. Blade row type for stator and rotor

Figure 36. Mesh 3D Solid View

4.1.1 Processing

After mesh generates, the next step is processing. Processing is done to known the performance of the turbine that has been designed. Before computer calculations begin, we need to set the parameters close to real conditions to get maximum results. **Figure 37.** is user interface of Numeca Fine Turbo.

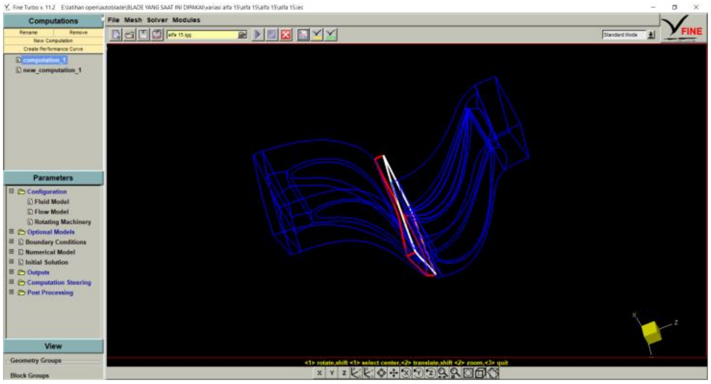

Figure 37. User Interface of NUMECA FINE Turbo

Figure After setting the parameters, then the calculation process by computer based on the boundary condition, shown in **Figure 38.** Inlet, and **Figure 39.** Outlet. All parameters can be seen on **Attachment 3.**

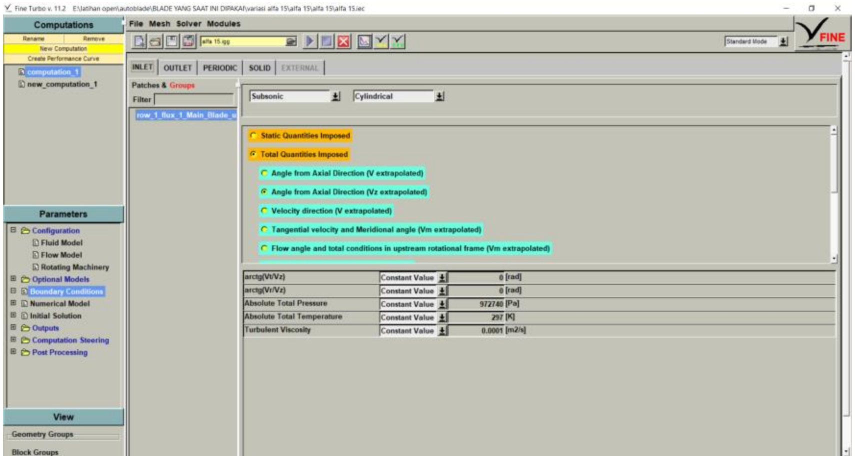

Figure 38. Boundary Condition Inlet

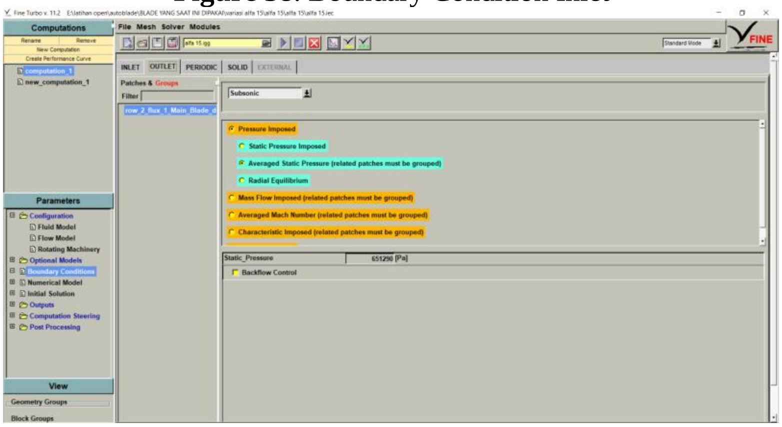

Figure 39. Boundary Condition Outlet

The iteration process is carried out until converge condition, mass flow in and out are in balanced condition. Mass flow graphic can be seen on **Figure 40.** efficiency graphic can be seen on **Figure 41.** Torque graphic can be seen on **Figure 42.** After reaching convergence conditions the results are summarized on the convergence history tab which can be seen on the **Figure 43**. Torque used to calculate power generated by OTEC Turbine. Mass flow used to calculate the warm and cold seawater requirement on evaporator and condenser.

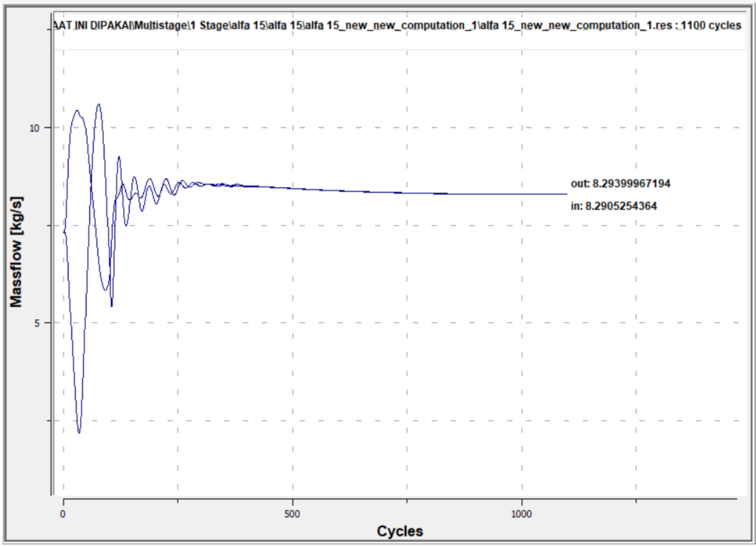

Figure 40. Mass Flow Graphic

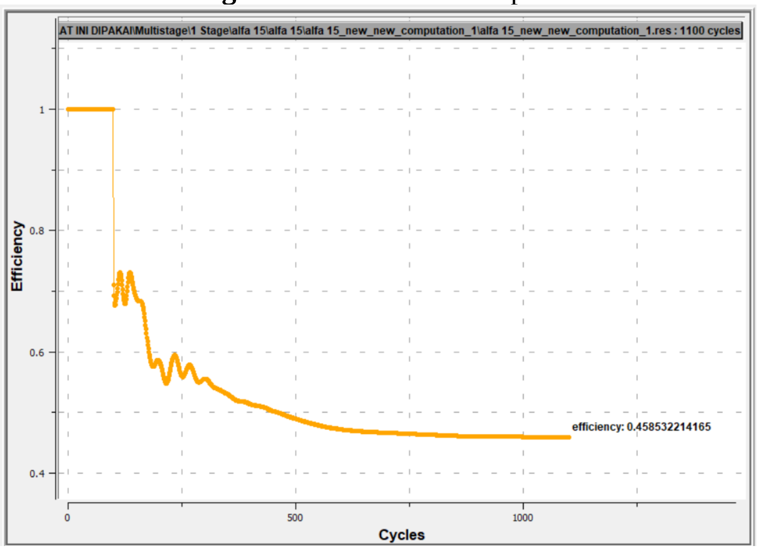

Figure 41. Efficiency Graphic

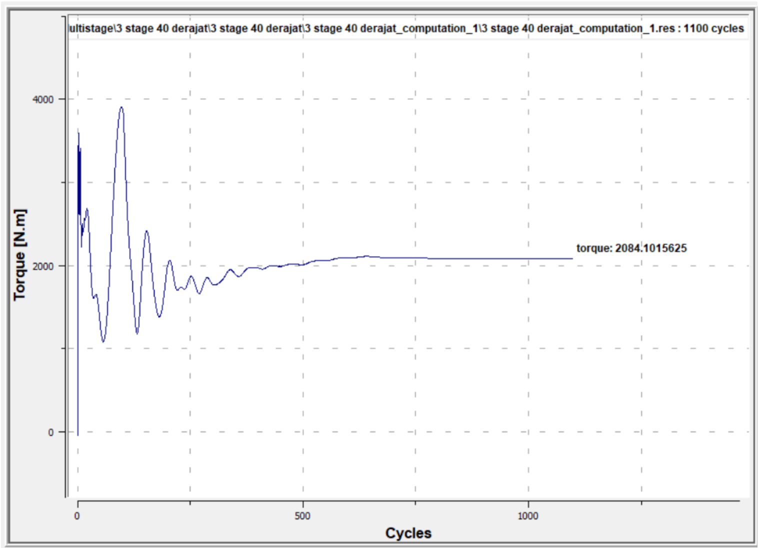

Figure 42. Torque Graphic

Selected Variables	Value	Units
global residual	-3.803e+000	
inlet mass flow	8.080e+000	[kg/s]
outlet mass flow	8.084e+000	[kg/s]
efficiency	4.660e-001	
pressure ratio	8.557e-001	
axial thrust	3.825e+003	[N]
torque	2.647e+002	[N.m]

Figure 43. Simulation Result

Simulation result shown in **Figure 43** such as mass flow, efficiency, and torque. Torque used to calculate power generated by OTEC Turbine. Mass flow used to calculate the warm and cold seawater requirement on evaporator and condenser. Convergency history for other turbine model can be seen on **Attachment 4.**

4.2 Performance of OTEC Turbine

In this research, will be simulated 1 model of single stage turbine, 11 model of turbine 2 stage with angle variation, and 1 model turbine 3 stage 40 degree. The model is simulated using Numeca Fine Turbo. The performance results of each turbine model will be compared to obtain the best OTEC turbine for laboratory scale. Simulation using Numeca Fine Turbo obtained numerical data, such as mass flow balance (inlet and outlet), efficiency, and torque shown in **Table 5**. Torque used to calculate power generated by OTEC Turbine. Mass flow used to calculate the warm and cold seawater needs on evaporator and condenser.

Table 5. Result of OTEC's Turbine Simulation

No.	Name	RPM	Mass Flow In (kg/s)	Mass Flow Out (kg/s)	Efficiency (%)	Torque
1	Single Stage	3000	8,291	8,294	45,85	288,8
2	2 Stage 0 Degree	3000	6,17	6,175	41,17	316,9
3	2 Stage 5 Degree	3000	8,21	8,19	42,3	500
4	2 Stage 10 Degree	3000	8,9	8,898	46,75	620,58
5	2 Stage 15 Degree	3000	12,37	12,38	45,33	958
6	2 Stage 20 Degree	3000	12,97	13,02	55,05	1177
7	2 Stage 25 Degree	3000	14,52	14,56	55,04	1396
8	2 Stage 30 Degree	3000	15,83	15,85	53,97	1584
9	2 Stage 35 Degree	3000	16,8	16,82	54,89	1744
10	2 Stage 40 Degree	3000	17,86	17,94	57,45	1943
11	2 Stage 45 Degree	3000	17,77	17,82	53,89	1852
12	2 Stage 50 Degree	3000	17,32	17,35	49,87	1632
13	3 Stage 40 Degree	3000	17,42	17,57	65,02	2084

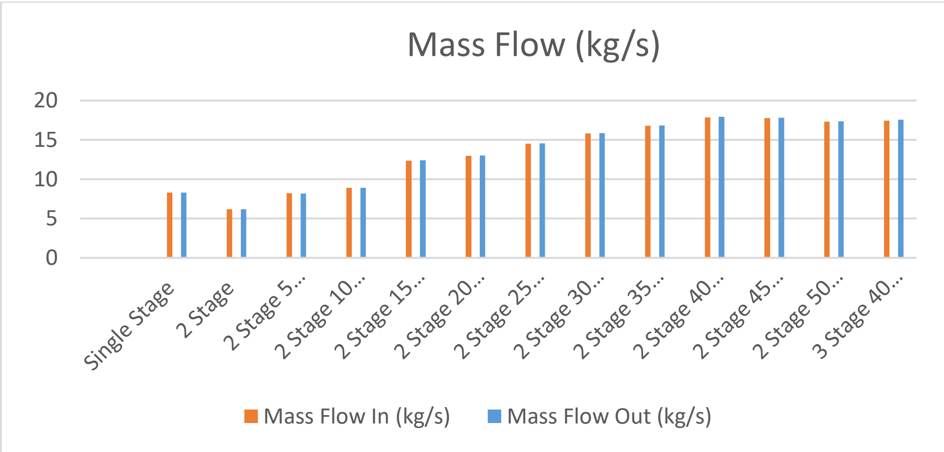

Figure 44. Mass Flow Error

There is a difference between mass flow in and mass flow out shown in **Figure 44**. The average error is less than 0.5%. The difference is relatively small so the calculations on the Numeca Fine Turbo are quite accurate.

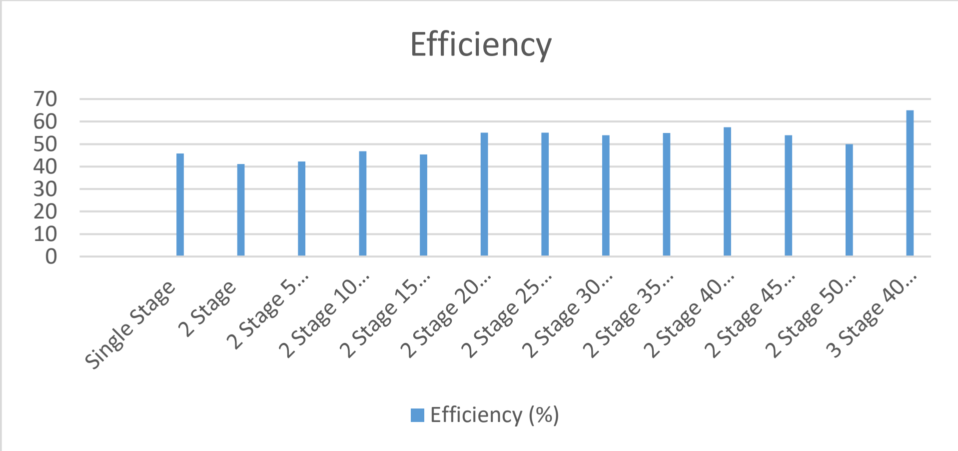

Figure 45. Efficiency of OTEC's Turbine

Figure 45 shown the efficiency from each model that has been simulated using Numeca Fine Turbo. Single stage turbine has an efficiency of 45.85%. For 2 stage turbine, the lowest efficiency is a straight turbine with the efficiency of 41.17%, and the highest efficiency is 2 stage turbine 40 degree with the efficiency of 57.45%. The 3 stage turbine has an efficiency of 65.02%. The author made a 3 stage turbine 40 degrees model in order to increase efficiency and power generated.

4.2.1 Calculation of OTEC Turbine Power

The results of the simulation using Numeca Fine Turbo is the torque that used to calculate the power generated by the OTEC turbine. The results of the simulation can be seen in **Table 5**. To known the power generated by the turbine can be calculated using the formula:

$$P = T \times 2\pi \times RPM/60000$$

Below is an example of turbine power calculation for a single stage turbine based on Table. In the same way, the result of turbine power shown in **Table 6**.

$$P = T \times 2\pi \times RPM/60000$$
$$P = 288.8 \times 2\left(\frac{22}{7}\right) \times \frac{3000}{60000}$$
$$P = 90.77\ kW$$

Table 6. Calculation of Turbine Power

No.	Name	RPM	Torque (Nm)	Power (kW)
1	Single Stage	3000	288,8	90,77
2	2 Stage	3000	316,9	99,60
3	2 Stage 5 Degree	3000	500	157,14
4	2 Stage 10 Degree	3000	620,58	195,04
5	2 Stage 15 Degree	3000	958	301,09
6	2 Stage 20 Degree	3000	1177	369,91
7	2 Stage 25 Degree	3000	1396	438,74
8	2 Stage 30 Degree	3000	1584	497,83
9	2 Stage 35 Degree	3000	1744	548,11
10	2 Stage 40 Degree	3000	1943	610,66
11	2 Stage 45 Degree	3000	1852	582,06
12	2 Stage 50 Degree	3000	1632	512,91
13	3 Stage 40 Degree	3000	2084	654,97

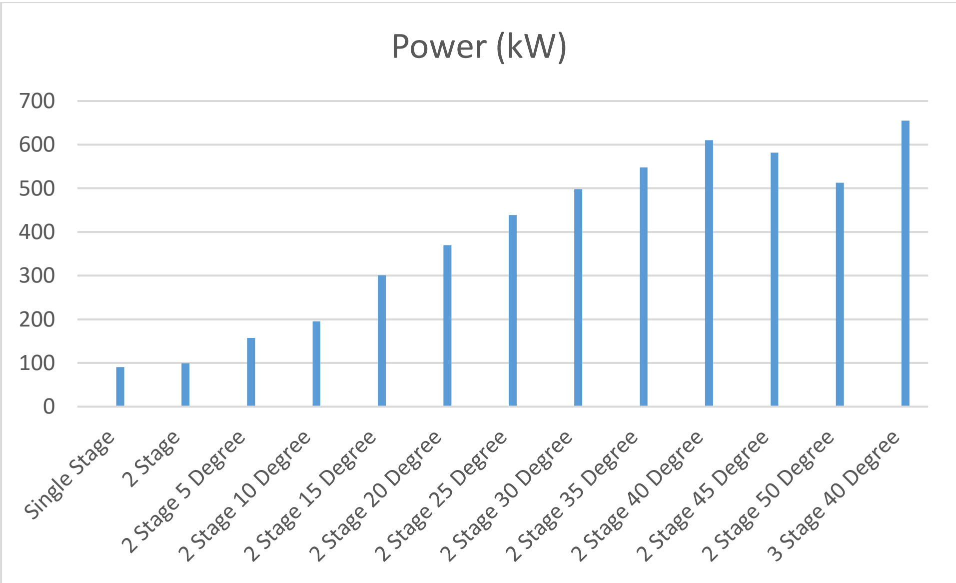

Figure 46. Efficiency of OTEC's Turbine

From **Figure 46** can be seen the power generated from each model that has been simulated using Numeca Fine Turbo. The single stage turbine produces 90.77 kW. Meanwhile, for the 2 stage turbine, the lowest is a straight turbine that produces 99.60 kW, and the highest is a turbine with 40 degree of slope that produces 610.66 kW. The author made a 3 stage turbine 40 degrees model in order to increase efficiency and power generated which is have 65.02% of efficiency, and produce 654.97 kW.

4.2.2 Calculation of Volume Flow Rate

In order for an Ocean Thermal Energy Conversion (OTEC) system to work, it needs ammonia as a working fluid, warm seawater used to change the ammonia phase from liquid to vapor, and cold seawater used to change the vapor phase to liquid. In this section will explain how to calculate the volume flow rate of ammonia, warm seawater, and cold seawater.

1. Calculation of Ammonia Volume Rate

   In the Ocean Thermal Energy Conversion (OTEC) system, ammonia is used as a working fluid that will rotate the turbine so that the turbine can generate power. simulation results in Table 6 has been known mass flow of ammonia. The following is an example of calculating of ammonia volume flow rate needs on a single stage turbine:

   $$Volume\ Flow\ Rate\ (Q) = \dot{m} \times v$$
   $$Volume\ Flow\ Rate\ (Q) = 8.297\ kg/s \times 1.6008 \times 10^{-3} m^3/kg$$
   $$Volume\ Flow\ Rate\ (Q) = 0.0013\ m^3/s$$
   $$Volume\ Flow\ Rate\ (Q) = 47.78\ m^3/h$$

   By using the same step, the result of ammonia volume flow rate can be seen in **Table 7**.

Table 7. Result of Ammonia Volume Flow Rate Calculation

No.	Name	Mass Flow In (kg/s)	Volume Flow Rate	
			m^3/s	(m^3/h)
1	Single Stage	8,291	0,0133	47,78
2	2 Stage	6,17	0,0099	35,56
3	2 Stage 5 Degree	8,21	0,0131	47,31
4	2 Stage 10 Degree	8,9	0,0142	51,29
5	2 Stage 15 Degree	12,37	0,0198	71,29
6	2 Stage 20 Degree	12,97	0,0208	74,74
7	2 Stage 25 Degree	14,52	0,0232	83,68
8	2 Stage 30 Degree	15,83	0,0253	91,23
9	2 Stage 35 Degree	16,8	0,0269	96,82
10	2 Stage 40 Degree	17,86	0,0286	102,93
11	2 Stage 45 Degree	17,77	0,0284	102,41

12	2 Stage 50 Degree	17,32	0,0277	99,81
13	3 Stage 40 Degree	17,42	0,0279	100,39

2. Calculation of Warm Seawater Volume Rate

To calculate the volume flow rate of warm seawater in the evaporator, it can be calculated by energy balance equation. The following is an example of calculating of warm seawater volume flow rate needs on a single stage turbine:

$Q_{ammonia} = Q_{Warm}$

$\dot{m}_{NH3} \times C_{pNH3} \times \Delta T = \dot{m}_{Warm} \times C_{pWarm} \times \Delta T$

$8.291 \times 2.026 \times 15 = \dot{m}_{Warm} \times 4.18 \times 3.55$

$251.96 = \dot{m}_{Warm} \times 14.8$

$\dot{m}_{Warm} = 16.98\ kg/s$

$Rate = \dot{m}_{Warm}/\rho$

$Rate = 16.98\ kg/s\ /\ 1025\ kg/m^3$

$Rate = 0.0166\ m^3/s = 59.64\ m^3/h$

By using the same step, the result of warm seawater volume flow rate can be seen in **Table 8**.

Table 8. Result of Warm Seawater Volume Flow Rate Calculation

No.	Name	Mass Flow (kg/s)	Density (kg/m3)	Volume Flow Rate	
				m3/s	m3/h
1	Single Stage	16,9798	1025	0,0166	59,6364
2	2 Stage	12,636	1025	0,0123	44,3803
3	2 Stage 5 Degree	16,8139	1025	0,0164	59,0538
4	2 Stage 10 Degree	18,227	1025	0,0178	64,0169
5	2 Stage 15 Degree	25,3335	1025	0,0247	88,9763
6	2 Stage 20 Degree	26,5623	1025	0,0259	93,2921
7	2 Stage 25 Degree	29,7367	1025	0,0290	104,441
8	2 Stage 30 Degree	32,4195	1025	0,0316	113,864
9	2 Stage 35 Degree	34,4061	1025	0,0336	120,841
10	2 Stage 40 Degree	36,577	1025	0,0357	128,465
11	2 Stage 45 Degree	36,3926	1025	0,0355	127,818
12	2 Stage 50 Degree	35,471	1025	0,0346	124,581
13	3 Stage 40 Degree	35,6758	1025	0,0348	125,301

3. Calculation of Cold Seawater Volume Rate

To calculate the volume flow rate of cold seawater in the condenser, it can be calculated by energy balance equation. The following is an example of calculating of cold seawater volume flow rate needs on a single stage turbine:

$$Q_{ammonia} = Q_{Cold}$$

$$\dot{m}_{NH3} \times C_{pNH3} \times \Delta T = \dot{m}_{Cold} \times C_{pCold} \times \Delta T$$

$$8.291 \times 4.676 \times 1 = \dot{m}_{Cold} \times 3.987 \times 2$$

$$38.77 = \dot{m}_{Cold} \times 7.947$$

$$\dot{m}_{Cold} = 4.86\ kg/s$$

$$Rate = \dot{m}_{NH3}/\rho$$

$$Rate = 4.86\ kg/s\ /\ 1025\ kg/m^3$$

$$Rate = 0.0047\ m^3/s = 17.08\ m^3/h$$

By using the same step, the result of warm seawater volume flow rate can be seen in **Table 9**.

Table 9. Result of Cold Seawater Volume Flow Rate Calculation

No.	Name	Mass Flow (kg/s)	Density (kg/m3)	Volume Flow Rate	
				m3/s	m3/h
1	Single Stage	4,86	1025	0,0047	17,08
2	2 Stage	3,62	1025	0,0035	12,71
3	2 Stage 5 Degree	4,81	1025	0,0047	16,91
4	2 Stage 10 Degree	5,22	1025	0,0051	18,33
5	2 Stage 15 Degree	7,25	1025	0,0071	25,48
6	2 Stage 20 Degree	7,61	1025	0,0074	26,71
7	2 Stage 25 Degree	8,51	1025	0,0083	29,90
8	2 Stage 30 Degree	9,28	1025	0,0091	32,60
9	2 Stage 35 Degree	9,85	1025	0,0096	34,60
10	2 Stage 40 Degree	10,47	1025	0,0102	36,78
11	2 Stage 45 Degree	10,42	1025	0,0102	36,60
12	2 Stage 50 Degree	10,16	1025	0,0099	35,67
13	3 Stage 40 Degree	10,22	1025	0,0100	35,84

4.2.3 Calculation of Heat Exchanger

The ocean thermal energy conversion (OTEC) system requires evaporators and condenser. An evaporator used to turn the liquid form of a chemical substance such as water into its gaseous-form/vapor. A condenser used to condense a substance from its gaseous to its liquid state, by cooling it. In this section will explain the calculation of the surface area and the number of tubes required for the evaporator and condenser.

1. Evaporator

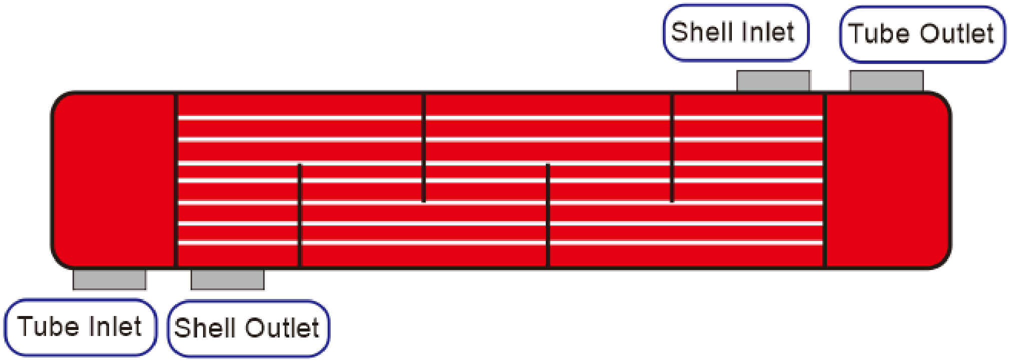

Figure 47. Evaporator

Figure 47 shown the Evaporator. In the evaporator, there is a difference of sea water temperature at inlet = 28°C and outlet = 24.45°C. In this research using tube and shell heat exchangers. Calculation using LMTD method. Below is an example calculation for evaporator on single stage turbine.

- Calculation of LMTD

$$LMTD = \frac{(T_{warm,in} - T_{NH3,out}) - (T_{warm,out} - T_{NH3,in})}{\ln\left[\frac{(T_{warm,in} - T_{NH3,out})}{(T_{warm,out} - T_{NH3,in})}\right]}$$

$$LMTD = \frac{(28°C - 24°C) - (24.5°C - 9°C)}{\ln\left[\frac{(28°C - 24°C)}{(24.5°C - 9°C)}\right]}$$

$$LMTD = 8.47$$

- LMTD Correction Factor (F)

  For shell and tube heat exchange cross-flow that having more than one or more multi flow passes, whether in shell or tube arrangement, in this case the LMTD values obtained must be corrected by correction factor (F). LMTD correction factor can be found by using

P and R parameters. P is the effectiveness of temperature on the cold fluid side. R is the ratio of heat energy capacity.

$$P = \frac{t_2 - t_1}{T_1 - t_1}$$

$$P = \frac{24°C - 9°C}{28°C - 9°C} = 0.789$$

$$R = \frac{T_1 - T_2}{t_2 - t_1}$$

$$R = \frac{28°C - 24.45°C}{24°C - 9°C} = 0.24$$

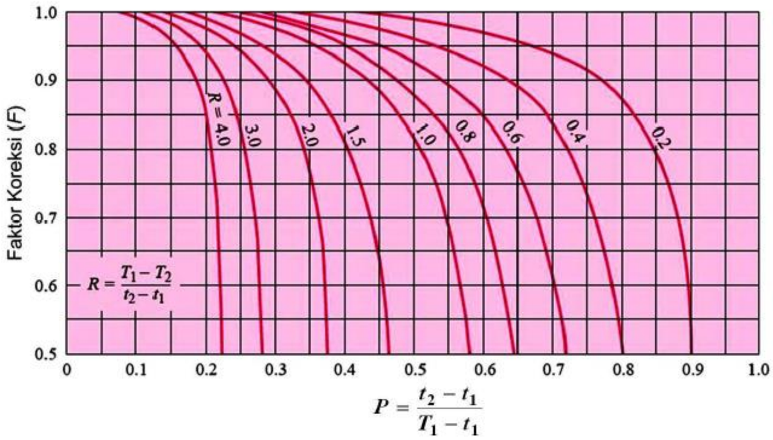

Figure 48. LMTD Correction Factor

Figure 48 shows the graph to determine LMTD correction factor, the LMTD correction factor is 0.89.

- Calculation of Evaporator's Cross-Sectional Area.
  In this research, the diameter and length of the pipes were 100 mm and 6000 mm. So, the evaporator cross-sectional area can be calculated using the equation:

$$A = \frac{Q}{U \times LMTD \times F}$$

$$A = \frac{251.96 \times 10^3 joule/s}{1000\ W/m^2.°C \times 8.47 \times 0.89}$$

$$A = 33.41\ m^2$$

- Calculation of number of tubes in evaporator

$$N_{tube} = \frac{A}{\pi \times D \times L}$$

$$N_{tube} = \frac{33.41}{\pi \times 0.1 \times 6}$$

$$N_{tube} = 17.7 = 18$$

By using the same step, the result of evaporator calculation can be seen in **Table 10**.

Table 10. Result of Evaporator Calculation

No.	Name	Evaporator							
		Q (kW)	LMTD	R	P	F	U (M/m2.C)	Area (m2)	N Pipe
1	Single Stage	251,963	8,47	0,24	0,79	0,89	1000	33,41	18
2	2 Stage	187,506	8,47	0,24	0,79	0,89	1000	24,86	14
3	2 Stage 5 Degree	249,502	8,47	0,24	0,79	0,89	1000	33,09	18
4	2 Stage 10 Degree	270,471	8,47	0,24	0,79	0,89	1000	35,87	20
5	2 Stage 15 Degree	375,924	8,47	0,24	0,79	0,89	1000	49,85	28
6	2 Stage 20 Degree	394,158	8,47	0,24	0,79	0,89	1000	52,27	28
7	2 Stage 25 Degree	441,263	8,47	0,24	0,79	0,89	1000	58,51	32
8	2 Stage 30 Degree	481,074	8,47	0,24	0,79	0,89	1000	63,79	34
9	2 Stage 35 Degree	510,552	8,47	0,24	0,79	0,89	1000	67,70	36
10	2 Stage 40 Degree	542,765	8,47	0,24	0,79	0,89	1000	71,97	38
11	2 Stage 45 Degree	540,03	8,47	0,24	0,79	0,89	1000	71,61	38
12	2 Stage 50 Degree	526,355	8,47	0,24	0,79	0,89	1000	69,80	38
13	3 Stage 50 Degree	529,394	8,47	0,24	0,79	0,89	1000	70,20	38

2. Condensor

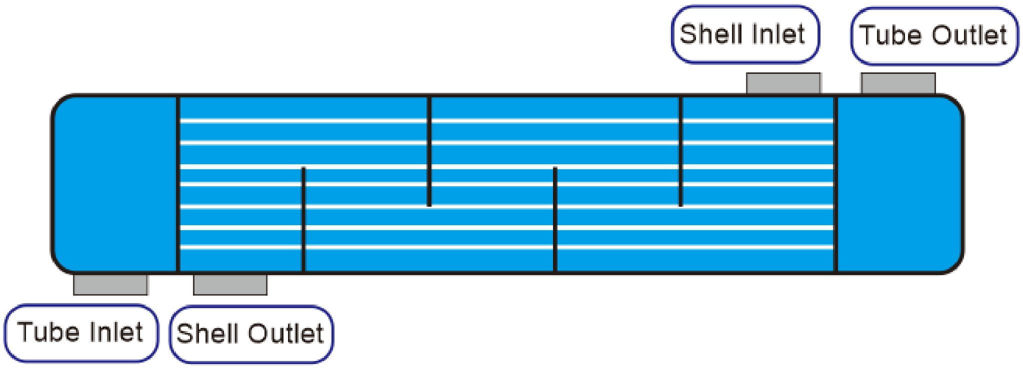

Figure 49. Condenser

Figure 49 shown the Condenser. In the condenser, there is a difference temperature of cold seawater temperature at inlet = 6°C and outlet = 8°C. In this research using tube and shell heat exchangers. Calculation using LMTD method. Below is an example calculation for condes on single stage turbine.

- Calculation of LMTD

$$LMTD = \frac{(T_{warm,in} - T_{NH3,out}) - (T_{warm,out} - T_{NH3,in})}{\ln\left[\frac{(T_{warm,in} - T_{NH3,out})}{(T_{warm,out} - T_{NH3,in})}\right]}$$

$$LMTD = \frac{(10°C - 8°C) - (9°C - 6°C)}{\ln\left[\frac{(10°C - 8°C)}{(9°C - 6°C)}\right]}$$

$$LMTD = 2.47$$

- LMTD Correction Factor (F)

For shell and tube heat exchange cross-flow that having more than one or more multi flow passes, whether in shell or tube arrangement, in this case the LMTD values obtained must be corrected by correction factor (F). LMTD correction factor can be found by using P and R parameters. P is the effectiveness of temperature on the cold fluid side. R is the ratio of heat energy capacity.

$$P = \frac{t_2 - t_1}{T_1 - t_1}$$

$$P = \frac{9°C - 10°C}{6°C - 10°C} = 0.25$$

$$R = \frac{T_1 - T_2}{t_2 - t_1}$$
$$R = \frac{6°C - 8°C}{9°C - 10°C} = 2$$

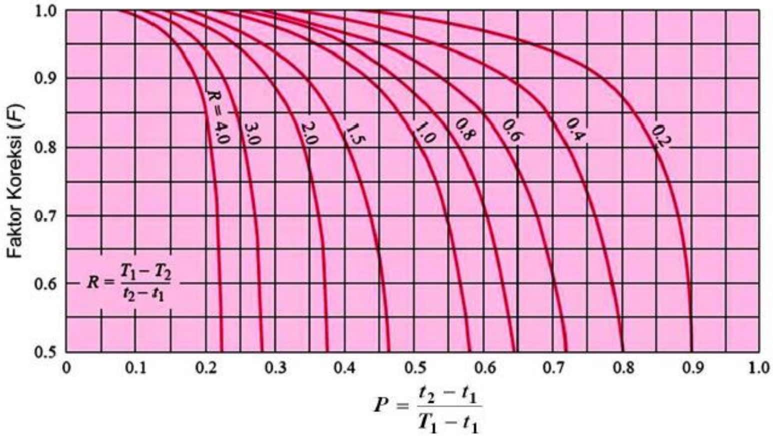

Figure 50. LMTD Correction Factor

Figure 50 shows the graph to determine LMTD correction factor, the LMTD correction factor is 0.89.

- Calculation of Evaporator's Cross-Sectional Area.
  In this research, the diameter and length of the pipes were 100 mm and 6000 mm. So, the evaporator cross-sectional area can be calculated using the equation:

$$A = \frac{Q}{U \times LMTD \times F}$$
$$A = \frac{38.77 \times 10^3 joule/s}{1000\ W/m^2.°C \times 2.47 \times 0.89}$$
$$A = 17.66\ m^2$$

- Calculation of number of tube in evaporator

$$N_{tube} = \frac{A}{\pi \times D \times L}$$
$$N_{tube} = \frac{17.66}{\pi \times 0.1 \times 6}$$
$$N_{tube} = 9.37 = 10$$

By using the same step, the result of evaporator calculation can be seen in **Table 11.**

Table 11. Result of Condenser Calculation

No.	Name	Condenser							
		Q (kW)	LMTD	R	P	F	U (M/m2.C)	Area (m2)	N Pipe
1	Single Stage	38,77	2,47	2,00	0,25	0,89	1000	17,66	10
2	2 Stage	28,85	2,47	2,00	0,25	0,89	1000	13,14	8
3	2 Stage 5 Degree	38,39	2,47	2,00	0,25	0,89	1000	17,49	10
4	2 Stage 10 Degree	41,62	2,47	2,00	0,25	0,89	1000	18,96	10
5	2 Stage 15 Degree	57,84	2,47	2,00	0,25	0,89	1000	26,35	14
6	2 Stage 20 Degree	60,65	2,47	2,00	0,25	0,89	1000	27,63	16
7	2 Stage 25 Degree	67,90	2,47	2,00	0,25	0,89	1000	30,93	18
8	2 Stage 30 Degree	74,02	2,47	2,00	0,25	0,89	1000	33,72	18
9	2 Stage 35 Degree	78,56	2,47	2,00	0,25	0,89	1000	35,79	20
10	2 Stage 40 Degree	83,51	2,47	2,00	0,25	0,89	1000	38,05	20
11	2 Stage 45 Degree	83,09	2,47	2,00	0,25	0,89	1000	37,86	20
12	2 Stage 50 Degree	80,99	2,47	2,00	0,25	0,89	1000	36,90	20
13	3 Stage 50 Degree	81,46	2,47	2,00	0,25	0,89	1000	37,11	20

4.2.4 Calculation of Pump

Ocean Thermal Energy Conversion (OTEC) system use about 3 pumps are used to pump the working fluid ie ammonia, warm seawater, and cold seawater. In this chapter will be explained about the calculation of the pump, including the calculation of head and power.

1. Working Fluid Pump

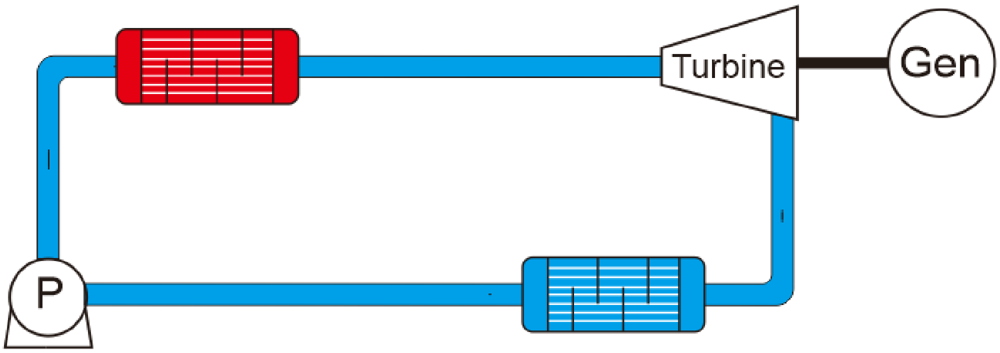

Figure 51. Working Fluid Pipeline

Figure 51 is a pipeline of the working fluid. The pump's total head should be provided for the planned amount of seawater. Ammonia needed is 47.78 m^3/h, the diameter of the pipe is 100 mm. so, the pump head can be calculated as follows:

1. Head static of pump (hs)
   Head static pump is calculated from pump inlet till the end of discharge. So, head static of pump (hs) = 2 m.

2. Head Pressure of pump (hp)
   Head Pressure is the diference of pressure on the suction and discharge. Because the pressure on th both side is same, sp thevalue of hp = 0 m.

3. Head Velocity of Pump (hv)
   Head Velocity (Hv) is difference velocity of fluid between in suction line in suction and discharge of pump. we can make assumption not difference between suction and discharge about flow velocity. So, head velocity of pump (hv) = 0 m.

4. Head Loss

Head losses are the sum of head major and minor on both of suction and discharge. Head Losses including head major and head minor in suction and discharge. Below is calculation example of single stage turbine.

4.1. Calculation of Suction Head

- Major Head Losses

$$Reynold\ number\ (Rn) = \frac{v \times d}{\mu}$$

$$Reynold\ number\ (Rn) = \frac{0.024 \times 0.1}{0.00000098} = 6925.7$$

Where: v = flow velocity (0.024 m/s)
d = inner diameter of pipe (0.1m)
μ = kinematis viscosity (9.8×10^{-7} m^2/s)

If Re>2300 = Turbulen
If Re<2300 = Laminer
Because of turbulen flow, so the equation

$$f = 0.02 + 0.0005/d$$

$$f = 0.025$$

$$Major\ losses = \frac{f \times L \times v^2}{d \times 2g}$$

Where:
L = Length of suction pipe = 60 m (estimation)
G = Gravitation = 9.8 m^2/s
D = inner diameter of pipe = 100 mm = 0.1 m

So, $Major\ losses = \frac{0.025 \times 60 \times 0.067^2}{0.1 \times 2(9.8)} = 0.0035\ m$

- Minor Head Loss

No	Accessories	n	k	n x k
1	Elbow 90°	2	0,8	1,6
2	Strainer	2	2,5	5
3	Gate Valve	2	0,2	0,4
Σ nk				7

$$Minor\ Losses = \frac{\Sigma nk \times v^2}{2g}$$

$$Minor\ Losses = \frac{7 \times 0.067^2}{2(9.8)} = 0.00158\ m$$

Head losses suction pipe = Major Loss + Minor Loss
= 0.0035 m + 0.00158 m
= 0.00508 m

4.2. Calculation of Discharge Head

- Major Head Losses

$$Major\ losses = \frac{f \times L \times v^2}{d \times 2g}$$

Where:
L = Length of suction pipe = 60 m (estimation)
G = Gravitation = 9.8 m^2/s
D = inner diameter of pipe = 00 mm = 0.1 m

So, $Major\ losses = \frac{0.025 \times 60 \times 0.067^2}{0.1 \times 2(9.8)} = 0.0035\ m$

- Minor Head Loss

No	Accessories	n	k	n x k
1	Elbow 90°	2	0,8	1,6
2	Strainer	2	2,5	5
3	Gate Valve	2	0,2	0,4
S nk				7

$$Minor\ Losses = \frac{\Sigma nk \times v^2}{2g}$$

$$Minor\ Losses = \frac{7 \times 0.067^2}{2(9.8)} = 0.0016\ m$$

Head losses suction pipe = Major Loss + Minor Loss
= 0.0035 m + 0.0016 m
= 0.0051 m

So, total head (H) = hs + hp + hv + hl
= 2.0051 m

After calculating pump head, then the power gained by the fluid can be calculated using equation:

$$P = \rho \times Q \times g \times H_{total}$$

$$P = 624.69 \text{ kg/m}^3 \times 0.013272 \text{ m}^3/\text{s} \times 9.8\text{m}/s^2 \times 2.0102\text{m}$$

$$P = 163.5\ Watt = 0.1635\ kW$$

After calculating pump head, then the power gained by the shaft can be calculated using equation:

$$P_{Pump} = \frac{P_{evap}}{\eta_{pump}}$$

$$P_{Pump} = \frac{0.1635\text{kW}}{75\%} = 0.22\ kW$$

In the same way, the result of pump power for other models can be seen in the **Table 12**.

Table 12. Calculation of Working Fluid Pump

No.	Name	Head Total (m)	Power		Shaft Power (kW)
			Watt	kW	
1	Single Stage	2,010156	163,4908	0,16	0,22
2	2 Stage	2,005624	121,3924	0,12	0,16
3	2 Stage 5 Degree	2,009958	161,8777	0,16	0,22
4	2 Stage 10 Degree	2,011702	175,6348	0,18	0,23
5	2 Stage 15 Degree	2,022607	245,4358	0,25	0,33
6	2 Stage 20 Degree	2,024853	257,6263	0,26	0,34
7	2 Stage 25 Degree	2,031148	289,311	0,29	0,39
8	2 Stage 30 Degree	2,037022	316,3249	0,32	0,42
9	2 Stage 35 Degree	2,041698	336,4786	0,34	0,45
10	2 Stage 40 Degree	2,047126	358,6598	0,36	0,48
11	2 Stage 45 Degree	2,046652	356,7699	0,36	0,48
12	2 Stage 50 Degree	2,044319	347,3388	0,35	0,46
13	3 Stage 50 Degree	2,044832	349,4319	0,35	0,47

2. Evaporator

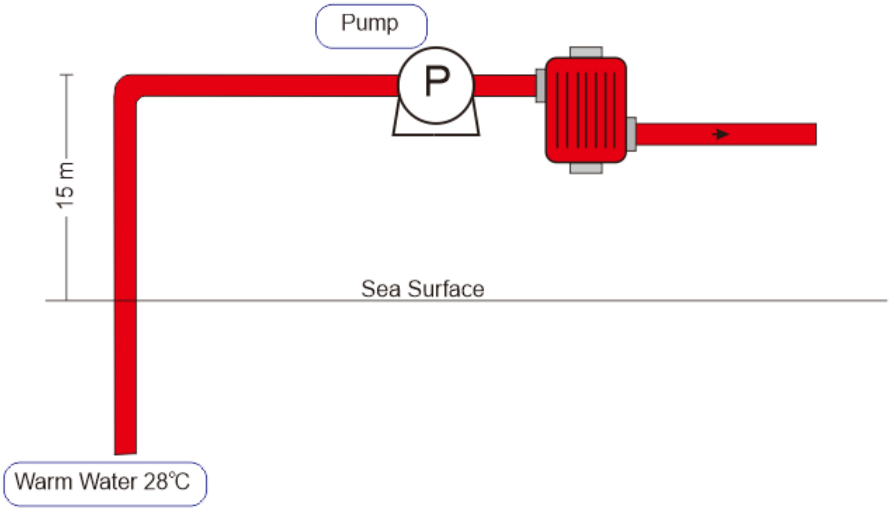

Figure 52. Evaporator Pipeline

Figure 52 is a pipeline of the evaporator. The pump's total head should be provided for the planned amount of seawater. Warm seawater needed is 59.64 m^3/h, the diameter of the pipe is 500 mm. Total head is the sum of head static, head pressure, Head Velocity, and head losses. So, the pump head can be calculated as follows:

1. Head static of pump (hs)
   Head static pump is calculated from pump inlet till the end of discharge. So, head static of pump (hs) = 15 m.

2. Head Pressure of pump (hp)
   Head Pressure is the diference of pressure on the suction and discharge. Because the pressure on the both side is same, the value of hp = 0 m.

3. Head Velocity of Pump (hv)
   Head Velocity (Hv) is difference velocity of fluid between in suction line in suction and discharge of pump. we can make assumption not difference between suction and discharge about flow velocity. So, head velocity of pump (hv) = 0 m.

4. Head Loss

Head losses are the sum of head major and minor on both of suction and discharge. Head Losses including head major and head minor in suction and discharge. Below is calculation example of single stage turbine.

4.1. Calculation of Suction Head

- Major Head Losses

$$Reynold\ number\ (Rn) = \frac{v \times d}{\mu}$$

$$Reynold\ number\ (Rn) = \frac{0.084334 \times 0.5}{0.00000098} = 43221.3$$

Where: v = flow velocity (0.084 m/s)
d = inner diameter of pipe (0.5m)
μ = kinematis viscosity (9.8 x 10^{-7} m^2/s)

If Re>2300 = Turbulen
If Re<2300 = Laminer
Because of turbulen flow, so the equation

$$f = 0.02 + 0.0005/d$$

$$f = 0.021$$

$$Major\ losses = \frac{f \times L \times v^2}{d \times 2g}$$

Where:
L = Length of suction pipe = 25 m (estimation)
G = Gravitation = 9.8 m^2/s
D = inner diameter of pipe = 500 mm = 0.5 m

So, $Major\ losses = \frac{0.021 \times 25 \times 0.084^2}{0.5 \times 2(9.8)} = 0.0004\ m$

- Minor Head Loss

No	Accessories	n	k	n x k
1	Elbow 90°	1	0,8	0,8
2	Strainer	1	2,5	2,5
3	Gate Valve	1	0,2	0,2
	Σ nk			3,4

$$Minor\ Losses = \frac{\Sigma nk \times v^2}{2g}$$

$$Minor\ Losses = \frac{3.4 \times 0.084^2}{2(9.8)} = 0.0012\ m$$

Head losses suction pipe = Major Loss + Minor Loss
= 0.0004 m + 0.0012 m
= 0.00016 m

b. Calculation of Discharge Head

- Major Head Losses

$$Major\ losses = \frac{f \times L \times v^2}{d \times 2g}$$

Where:
L = Length of suction pipe = 15 m (estimation)
G = Gravitation = 9.8 m^2/s
D = inner diameter of pipe = 500 mm = 0.5 m

So, $Major\ losses = \frac{0.021 \times 15 \times 0.084^2}{0.5 \times 2(9.8)} = 0.00023\ m$

- Minor Head Loss

No	Accessories	n	k	n x k
1	Elbow 90°	1	0,8	0,8
2	Strainer	1	2,5	2,5
3	Gate Valve	1	0,2	0,2
S nk				3,4

$$Minor\ Losses = \frac{\Sigma nk \times v^2}{2g}$$

$$Minor\ Losses = \frac{3.4 \times 0.084^2}{2(9.8)} = 0.0012\ m$$

Head losses suction pipe = Major Loss + Minor Loss
= 0.00023 m + 0.00123 m
= 0.00146 m

So, total head (H) = hs + hp + hv + hl
= 15.0031 m

After calculating pump head, then the power gained by the fluid can be calculated using equation:

$$P = \rho \times Q \times g \times H_{total}$$
$$P = 1025\ \mathrm{kg/m^3} \times 0.017\ \mathrm{m^3/s} \times 9.8 \mathrm{m}/s^2 \times 15.0031\mathrm{m}$$
$$P = 2499.09\ Watt = 2.5\ kW$$

After calculating pump head, then the power gained by the shaft can be calculated using equation:

$$P_{Pump} = \frac{P_{evap}}{\eta_{pump}}$$

$$P_{Pump} = \frac{2.5\mathrm{kW}}{75\%} = 3.33\ kW$$

In the same way, the result of pump power for other models can be seen in the **Table 13**.

Table 13. Calculation of Evaporator Pump

No.	Name	Rate (m^3/h)	Head Total (m)	Power (kW)
1	Single Stage	59,64	15,0031	3,33
2	2 Stage	44,38	15,0017	2,48
3	2 Stage 5 Degree	59,05	15,003	3,30
4	2 Stage 10 Degree	64,02	15,0035	3,58
5	2 Stage 15 Degree	88,98	15,0068	4,97
6	2 Stage 20 Degree	93,29	15,0075	5,21
7	2 Stage 25 Degree	104,44	15,0094	5,84
8	2 Stage 30 Degree	113,86	15,0112	6,37
9	2 Stage 35 Degree	120,84	15,0126	6,76
10	2 Stage 40 Degree	128,47	15,0143	7,18
11	2 Stage 45 Degree	127,82	15,0141	7,15
12	2 Stage 50 Degree	124,58	15,0134	6,97
13	3 Stage 50 Degree	125,30	15,0136	7,01

5. Condenser

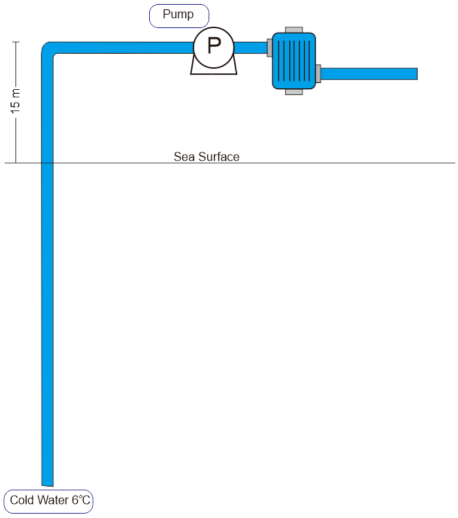

Figure 53. Condenser Pipeline

Figure 53 is a pipeline of the condenser. The pump's total head should be provided for the planned amount of seawater. Cold seawater needed is 17.08 m^3/h, the diameter of the pipe is 500 mm. Total head is the sum of head static, head pressure, Head Velocity, and head losses. So, the pump head can be calculated as follows:

1. Head static of pump (hs)
   Head static pump is calculated from pump inlet till the end of discharge. So, head static of pump (hs) = 15 m.

2. Head Pressure of pump (hp)
   Head Pressure is the diference of pressure on the suction and discharge. Because the pressure on th both side is same, sp thevalue of hp = 0 m.

3. Head Velocity of Pump (hv)
   Head Velocity (Hv) is difference velocity of fluid between in suction line in suction and discharge of pump. we can make assumption not difference between suction and discharge about flow velocity. So, head velocity of pump (hv) = 0 m.

4. Head Losses

Head losses are the sum of head major and minor on both of suction and discharge. Head Losses including head major and head minor in suction and discharge. Below is calculation example of single stage turbine.

4.1. Calculation of Suction Head

- Major Head Losses

$$Reynold\ number\ (Rn) = \frac{v \times d}{\mu}$$

$$Reynold\ number\ (Rn) = \frac{0.024 \times 0.5}{0.00000098} = 12375.7$$

Where: v = flow velocity (0.024 m/s)
d = inner diameter of pipe (0.5m)
μ = kinematis viscosity (9.8 x 10^{-7} m^2/s)

If Re>2300 = Turbulen
If Re<2300 = Laminer
Because of turbulen flow, so the equation

$$f = 0.02 + 0.0005/d$$

$$f = 0.021$$

$$Major\ losses = \frac{f \times L \times v^2}{d \times 2g}$$

Where:
L = Length of suction pipe = 650 m (estimation)
G = Gravitation = 9.8 m^2/s
D = inner diameter of pipe = 500 mm = 0.5 m

So, $Major\ losses = \frac{0.021 \times 650 \times 0.024^2}{0.5 \times 2(9.8)} = 0.0007\ m$

- Minor Head Loss

No	Accessories	n	k	n x k
1	Elbow 90°	1	0,8	0,8
2	Strainer	1	2,5	2,5
3	Gate Valve	1	0,2	0,2
	Σ nk			3,4

$$Minor\ Losses = \frac{\Sigma nk \times v^2}{2g}$$

$$Minor\ Losses = \frac{3.4 \times 0.024^2}{2(9.8)} = 0.0001\ m$$

Head losses suction pipe = Major Loss + Minor Loss
= 0.0007 m + 0.0001 m
= 0.0008 m

4.2. Calculation of Discharge Head

- Major Head Losses

$$Major\ losses = \frac{f \times L \times v^2}{d \times 2g}$$

Where:
L = Length of suction pipe = 15 m (estimation)
G = Gravitation = 9.8 m^2/s
D = inner diameter of pipe = 500 mm = 0.5 m

So, $Major\ losses = \frac{0.021 \times 15 \times 0.024^2}{0.5 \times 2(9.8)} = 0.00001\ m$

- Minor Head Loss

No	Accessories	n	k	n x k
1	Elbow 90^o	1	0,8	0,8
2	Strainer	1	2,5	2,5
3	Gate Valve	1	0,2	0,2
	S nk			3,4

$$Minor\ Losses = \frac{\Sigma nk \times v^2}{2g}$$

$$Minor\ Losses = \frac{3.4 \times 0.024^2}{2(9.8)} = 0.0001\ m$$

Head losses suction pipe = Major Loss + Minor Loss
= 0.00001 m + 0.0001 m
= 0.00011 m

So, total head (H) = hs + hp + hv + hl
= 15.001 m

After calculating pump head, then the power gained by the fluid can be calculated using equation:

$$P = \rho \times Q \times g \times H_{total}$$
$$P = 1025\ \mathrm{kg/m^3} \times 0.0047\ \mathrm{m^3/s} \times 9.8\mathrm{m}/s^2 \times 15.001\mathrm{m}$$
$$P = 715.47\ Watt = 0.72\ kW$$

After calculating pump head, then the power gained by the shaft can be calculated using equation:

$$P_{Pump} = \frac{P_{evap}}{\eta_{pump}}$$

$$P_{Pump} = \frac{0.72\mathrm{kW}}{75\%} = 0.95\ kW$$

In the same way, the result of pump power for other models can be seen in the **Table 14**.

Table 14. Calculation of Condenser Pump

No.	Name	Head Total (m)	Power		Shaft Power (kW)
			Watt	kW	
1	Single Stage	15,00096	715,4732	0,7	0,95
2	2 Stage	16,00053	1983,42	2,0	2,64
3	2 Stage 5 Degree	17,00095	2804,215	2,8	3,74
4	2 Stage 10 Degree	18,00111	3218,729	3,2	4,29
5	2 Stage 15 Degree	19,00215	4722,451	4,7	6,30
6	2 Stage 20 Degree	20,00236	5212,143	5,2	6,95
7	2 Stage 25 Degree	21,00296	6126,919	6,1	8,17
8	2 Stage 30 Degree	22,00352	6997,905	7,0	9,33
9	2 Stage 35 Degree	23,00396	7764,383	7,8	10,35
10	2 Stage 40 Degree	24,00448	8613,284	8,6	11,48
11	2 Stage 45 Degree	25,00443	8926,875	8,9	11,90
12	2 Stage 50 Degree	26,00421	9048,709	9,0	12,06
13	3 Stage 50 Degree	27,00425	9450,947	9,5	12,60

4.2.5 Calculation of Nett Power

Nett power is the power that distributed to the consumer. Generator is needed to generate power that will be distributed to the consumer. In this research, generator efficiency is assumed to be 85%. **Table 15.** shown the power generated by generator.

Table 15. Power generated by generator

No.	Name	Efficiency (%)	Turbine Power (kW)		Generator (kW)	
			Simulation	Nett	Efficiency	Power
1	Single Stage	45,85	90,77	41,62	85	35,37
2	2 Stage	41,17	99,60	41,00	85	34,85
3	2 Stage 5 Degree	42,3	157,14	66,47	85	56,50
4	2 Stage 10 Degree	46,75	195,04	91,18	85	77,50
5	2 Stage 15 Degree	45,33	301,09	136,48	85	116,01
6	2 Stage 20 Degree	55,05	369,91	203,64	85	173,09
7	2 Stage 25 Degree	55,04	438,74	241,48	85	205,26
8	2 Stage 30 Degree	53,97	497,83	268,68	85	228,38
9	2 Stage 35 Degree	54,89	548,11	300,86	85	255,73
10	2 Stage 40 Degree	57,45	610,66	350,82	85	298,20
11	2 Stage 45 Degree	53,89	582,06	313,67	85	266,62
12	2 Stage 50 Degree	49,87	512,91	255,79	85	217,42
13	3 Stage 40 Degree	65,02	654,97	425,86	85	361,98

After knowing the power generated by the generator, the pump calculation needs to be done to know the nett power to be distributed to the consumer. **Table 16.** shown nett power to be delivered to the consumers.

Table 16. Nett Power

No.	Name	Power by Generator (kW)	Pump Power (kW)			Nett Power (kW)
			Evap	Cond	NH3	
1	Single Stage	35,37	3,33	0,95	0,22	30,87
2	2 Stage	34,85	2,48	0,76	0,16	31,45
3	2 Stage 5 Degree	56,50	3,30	1,07	0,22	51,91
4	2 Stage 10 Degree	77,50	3,58	1,23	0,23	72,46
5	2 Stage 15 Degree	116,01	4,97	1,80	0,33	108,91

6	2 Stage 20 Degree	173,09	5,21	1,99	0,34	165,54
7	2 Stage 25 Degree	205,26	5,84	2,34	0,39	196,70
8	2 Stage 30 Degree	228,38	6,37	2,67	0,42	218,92
9	2 Stage 35 Degree	255,73	6,76	2,96	0,45	245,56
10	2 Stage 40 Degree	298,20	7,18	3,29	0,48	287,25
11	2 Stage 45 Degree	266,62	7,15	3,41	0,48	255,59
12	2 Stage 50 Degree	217,42	6,97	3,45	0,46	206,54
13	3 Stage 40 Degree	361,98	7,01	3,60	0,47	350,91

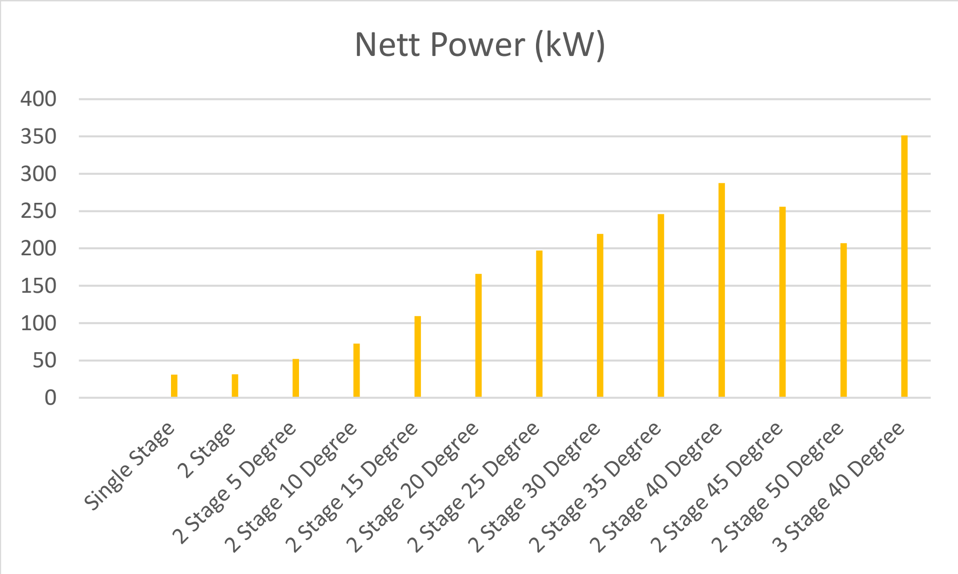

Figure 54. Nett Power of OTEC's Turbine

Figure 54. Can be seen that a single-stage OTEC turbine produces nett power of 31.09 kW. For the 2 stage OTEC turbine, the lowest power is a straight turbine model with a nett power of 31.62 kW, while the highest is generated by a 40 degree turbine with a nett power of 287.73 kW. For 3 stage OTEC turbines producing the highest power among all models, the resulting nett power is 351.37 kW.

4.3 Study Case

In this research conducted case studies to find out how the influence of mass flow to power and turbine efficiency. The turbine model taken in this study case is a 2 stage 10 degree turbine, then mass flow varied ie 5kg / s, 10kg / s, 15kg / s, and 20kg / s. The results of the simulation can be seen in table 17 and 18.

Table 17. Simulation Result

No.	Name	RPM	Mass Flow In (kg/s)	Mass Flow Out (kg/s)	Efficiency (%)
1	2 Stage 10 Degree	3000	5	5.033	76.9
2	2 Stage 10 Degree	3000	8.9	8.89	46.75
3	2 Stage 10 Degree	3000	10	9.958	45.19
4	2 Stage 10 Degree	3000	15	15	42.35
5	2 Stage 10 Degree	3000	20	20.01	41.69

Table 18. Turbine Power

No.	Name	Torque	Power (kW)	
			Simulation	Nett
1	2 Stage 10 Degree	207	65.05714	50.02894
2	2 Stage 10 Degree	620.58	195.0394	91.18093
3	2 Stage 10 Degree	763	239.8	108.3656
4	2 Stage 10 Degree	1457	457.9143	193.9267
5	2 Stage 10 Degree	2169	681.6857	284.1948

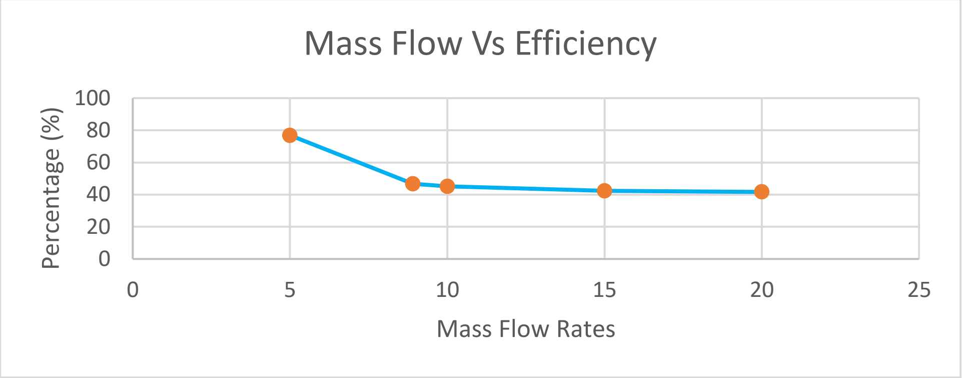

Figure 55. Mass Flow Vs Efficiency

From the graph in Figure 55 can be seen that 5kg / s mass flow has the highest efficiency of 76.9%, while the lowest efficiency is mass flow 20 kg/s with the efficiency of 41.69%.

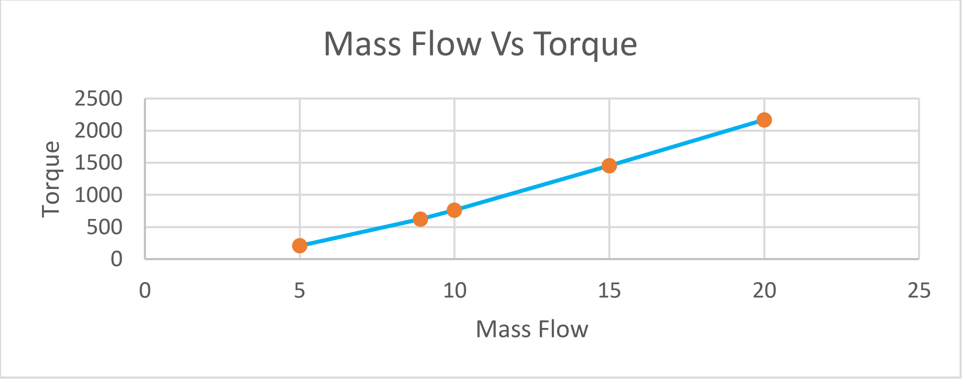

Figure 56. Mass Flow Vs Torque

From the graph in Figure 55 can be seen that 5kg / s mass flow has the lowest torque of 207 Nm, while the highest is mass flow 20 kg/s with the torque of 2169 Nm.

CHAPTER V
CONCLUSION

5.1 Conclusion

Based on the results of design and simulation that has been done, it can be concluded as follows:

1. Design of the OTEC turbine conducted based on thermodynamic conditions on a region. Turbine with T_{in} = 24 ° C, P_{in} = 9.7274 bar, and T_{out} = 10 ° C, P_{out} = 6,1529 bar, have an enthalpy difference of 28.6 kJ / kg using ammonia as working fluid. Result of calclation turbine blades angle with α_1 = 15° are: α_2 = 31.78°, β_1 = 24.1°, β_2 = 21.1°.

2. The highest efficiency and net power is a 3 stage 40 degree turbine with 351.37 kW generated power, and 65.02% efficiency. Lowest is single stage turbine with nett power 31.09 kW, and efficiency 45.85%.

5.2 Sugestion

1. Data collection in the field need to be done to get the result of research approaching with real situation.
2. Further researches should be conducted about factors that affect the performance of the OTEC turbine.
3. An experiment is needed to know the performance of OTEC turbines in real terms.

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Syamsuddin, M. L., Attamimi, A., Nugraha, A. P., Gibran, S., Afifah, A. Q., & Oriana, N. (2015). OTEC Potential in the Indonesian Seas. Energy Procedia, 65, 215–222. https://doi.org/10.1016/j.egypro.2015.01.028

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ATTACHMENT

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Attachment 1.
Table Properties of Ammonia

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Table Properties of Ammonia

948 Tables in SI Units

TABLE A-13

Pressure Conversions:
1 bar = 0.1 MPa
= 10^2 kPa

Properties of Saturated Ammonia (Liquid–Vapor): Temperature Table

		Specific Volume m^3/kg		Internal Energy kJ/kg		Enthalpy kJ/kg			Entropy kJ/kg · K		
Temp. °C	Press. bar	Sat. Liquid $v_f \times 10^3$	Sat. Vapor v_g	Sat. Liquid u_f	Sat. Vapor u_g	Sat. Liquid h_f	Evap. h_{fg}	Sat. Vapor h_g	Sat. Liquid s_f	Sat. Vapor s_g	Temp. °C
−50	0.4086	1.4245	2.6265	−43.94	1264.99	−43.88	1416.20	1372.32	−0.1922	6.1543	−50
−45	0.5453	1.4367	2.0060	−22.03	1271.19	−21.95	1402.52	1380.57	−0.0951	6.0523	−45
−40	0.7174	1.4493	1.5524	−0.10	1277.20	0.00	1388.56	1388.56	0.0000	5.9557	−40
−36	0.8850	1.4597	1.2757	17.47	1281.87	17.60	1377.17	1394.77	0.0747	5.8819	−36
−32	1.0832	1.4703	1.0561	35.09	1286.41	35.25	1365.55	1400.81	0.1484	5.8111	−32
−30	1.1950	1.4757	0.9634	43.93	1288.63	44.10	1359.65	1403.75	0.1849	5.7767	−30
−28	1.3159	1.4812	0.8803	52.78	1290.82	52.97	1353.68	1406.66	0.2212	5.7430	−28
−26	1.4465	1.4867	0.8056	61.65	1292.97	61.86	1347.65	1409.51	0.2572	5.7100	−26
−22	1.7390	1.4980	0.6780	79.46	1297.18	79.72	1335.36	1415.08	0.3287	5.6457	−22
−20	1.9019	1.5038	0.6233	88.40	1299.23	88.68	1329.10	1417.79	0.3642	5.6144	−20
−18	2.0769	1.5096	0.5739	97.36	1301.25	97.68	1322.77	1420.45	0.3994	5.5837	−18
−16	2.2644	1.5155	0.5291	106.36	1303.23	106.70	1316.35	1423.05	0.4346	5.5536	−16
−14	2.4652	1.5215	0.4885	115.37	1305.17	115.75	1309.86	1425.61	0.4695	5.5239	−14
−12	2.6798	1.5276	0.4516	124.42	1307.08	124.83	1303.28	1428.11	0.5043	5.4948	−12
−10	2.9089	1.5338	0.4180	133.50	1308.95	133.94	1296.61	1430.55	0.5389	5.4662	−10
−8	3.1532	1.5400	0.3874	142.60	1310.78	143.09	1289.86	1432.95	0.5734	5.4380	−8
−6	3.4134	1.5464	0.3595	151.74	1312.57	152.26	1283.02	1435.28	0.6077	5.4103	−6
−4	3.6901	1.5528	0.3340	160.88	1314.32	161.46	1276.10	1437.56	0.6418	5.3831	−4
−2	3.9842	1.5594	0.3106	170.07	1316.04	170.69	1269.08	1439.78	0.6759	5.3562	−2
0	4.2962	1.5660	0.2892	179.29	1317.71	179.96	1261.97	1441.94	0.7097	5.3298	0
2	4.6270	1.5727	0.2695	188.53	1319.34	189.26	1254.77	1444.03	0.7435	5.3038	2
4	4.9773	1.5796	0.2514	197.80	1320.92	198.59	1247.48	1446.07	0.7770	5.2781	4
6	5.3479	1.5866	0.2348	207.10	1322.47	207.95	1240.09	1448.04	0.8105	5.2529	6
8	5.7395	1.5936	0.2195	216.42	1323.96	217.34	1232.61	1449.94	0.8438	5.2279	8
10	6.1529	1.6008	0.2054	225.77	1325.42	226.75	1225.03	1451.78	0.8769	5.2033	10
12	6.5890	1.6081	0.1923	235.14	1326.82	236.20	1217.35	1453.55	0.9099	5.1791	12
16	7.5324	1.6231	0.1691	253.95	1329.48	255.18	1201.70	1456.87	0.9755	5.1314	16
20	8.5762	1.6386	0.1492	272.86	1331.94	274.26	1185.64	1459.90	1.0404	5.0849	20
24	9.7274	1.6547	0.1320	291.84	1334.19	293.45	1169.16	1462.61	1.1048	5.0394	24
28	10.993	1.6714	0.1172	310.92	1336.20	312.75	1152.24	1465.00	1.1686	4.9948	28
32	12.380	1.6887	0.1043	330.07	1337.97	332.17	1134.87	1467.03	1.2319	4.9509	32
36	13.896	1.7068	0.0930	349.32	1339.47	351.69	1117.00	1468.70	1.2946	4.9078	36
40	15.549	1.7256	0.0831	368.67	1340.70	371.35	1098.62	1469.97	1.3569	4.8652	40
45	17.819	1.7503	0.0725	393.01	1341.81	396.13	1074.84	1470.96	1.4341	4.8125	45
50	20.331	1.7765	0.0634	417.56	1342.42	421.17	1050.09	1471.26	1.5109	4.7604	50

v_f = (table value)/1000

Ammonia

Figure 1. Table Properties of Ammonia

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Attachment 2.
OTEC Turbine Models

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OTEC Turbine Models

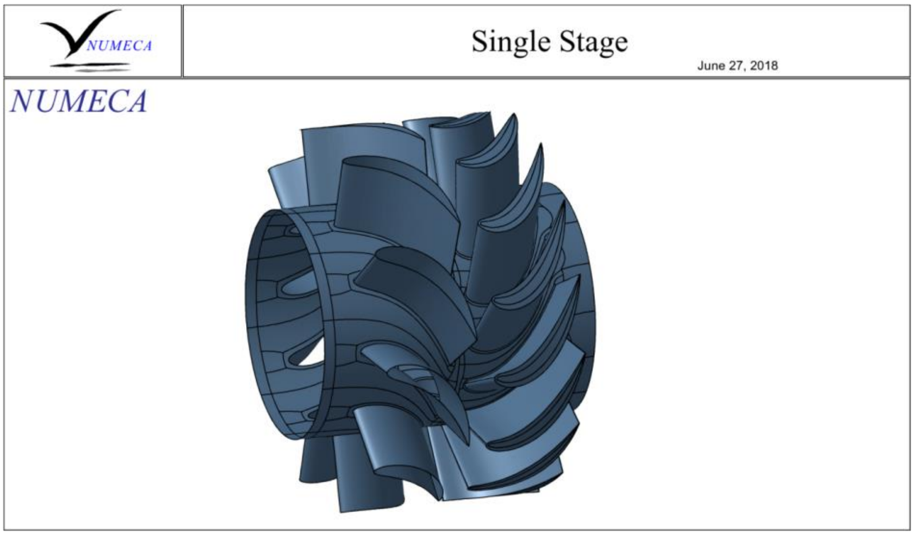

Figure Single Stage Turbine

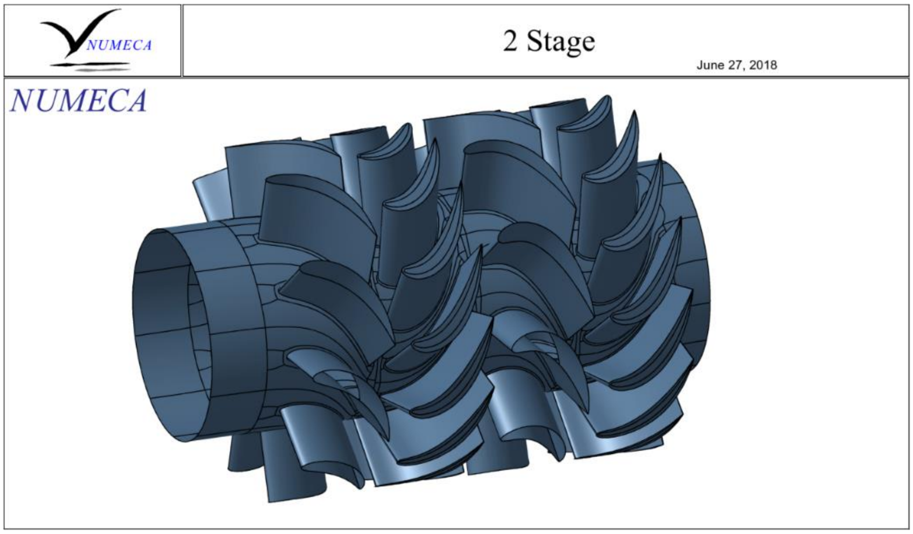

Figure 2 Stage Turbine

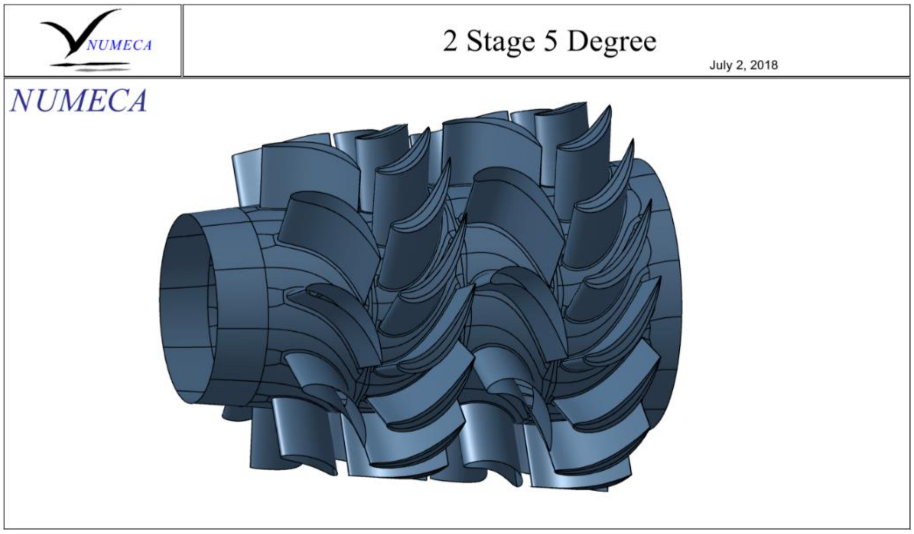

Figure 3. 2 Stage 5 Degree Turbine

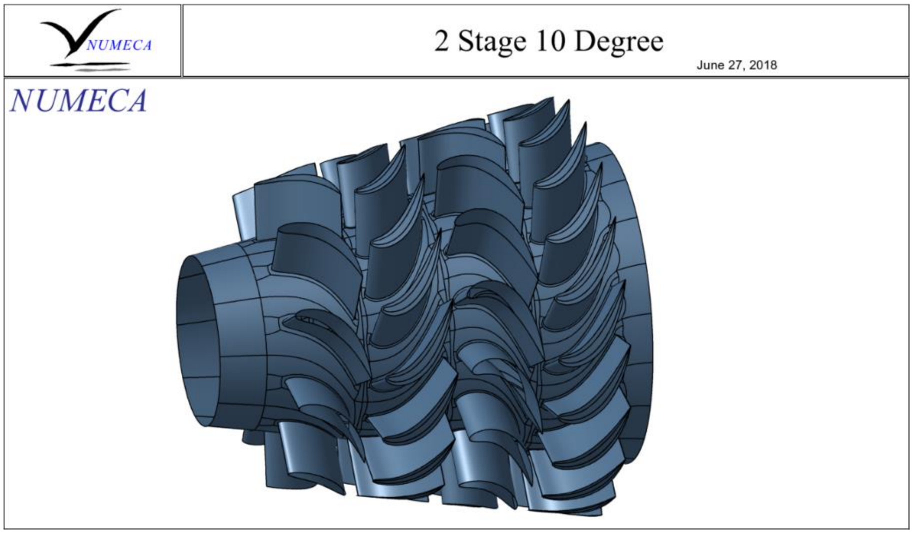

Figure 4. 2 Stage 10 Degree Turbine

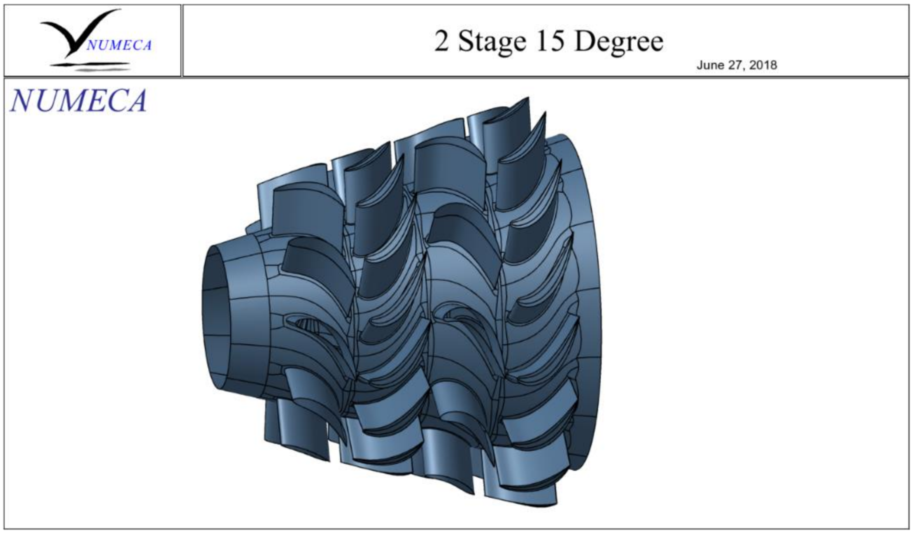

Figure 5. 2 Stage 15 Degree Turbine

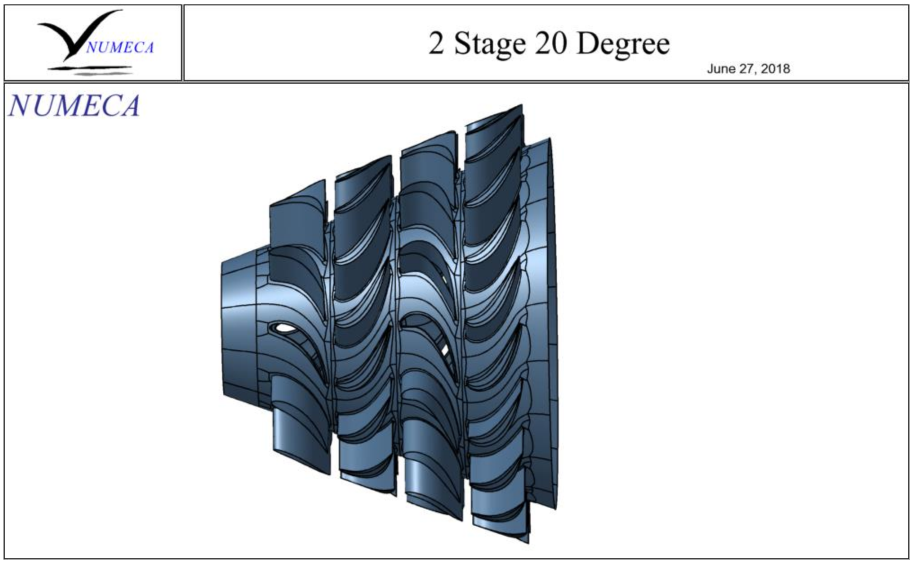

Figure 6. 2 Stage 20 Degree Turbine

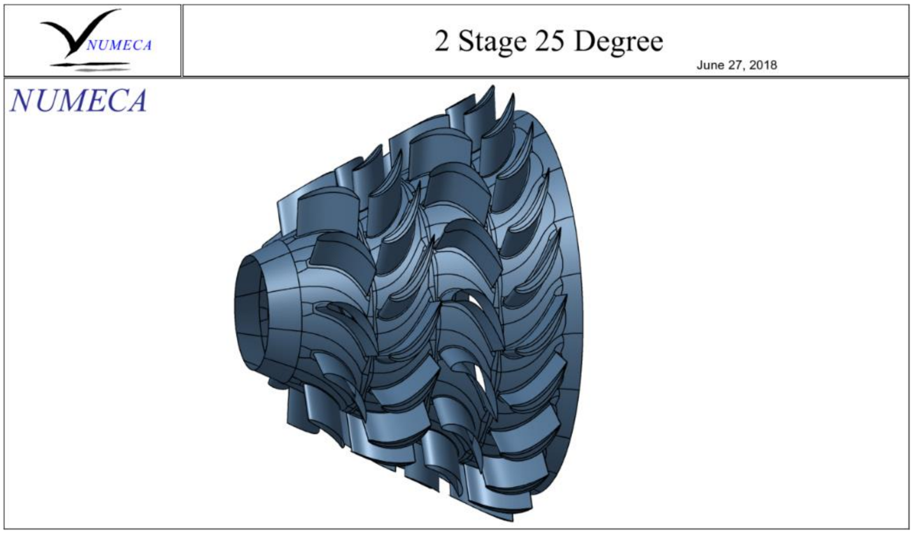

Figure 7. 2 Stage 25 Degree Turbine

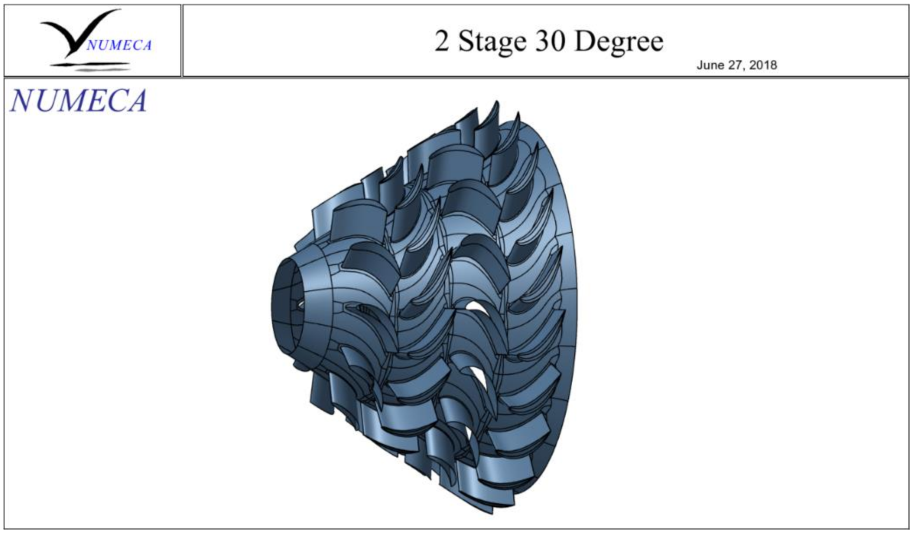

Figure 8. 2 Stage 30 Degree Turbine

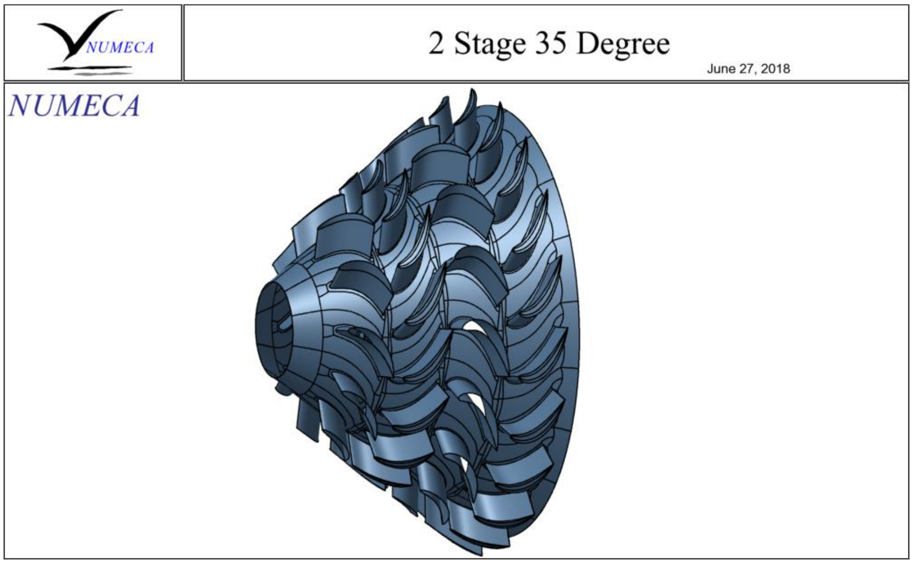

Figure 9. 2 Stage 35 Degree Turbine

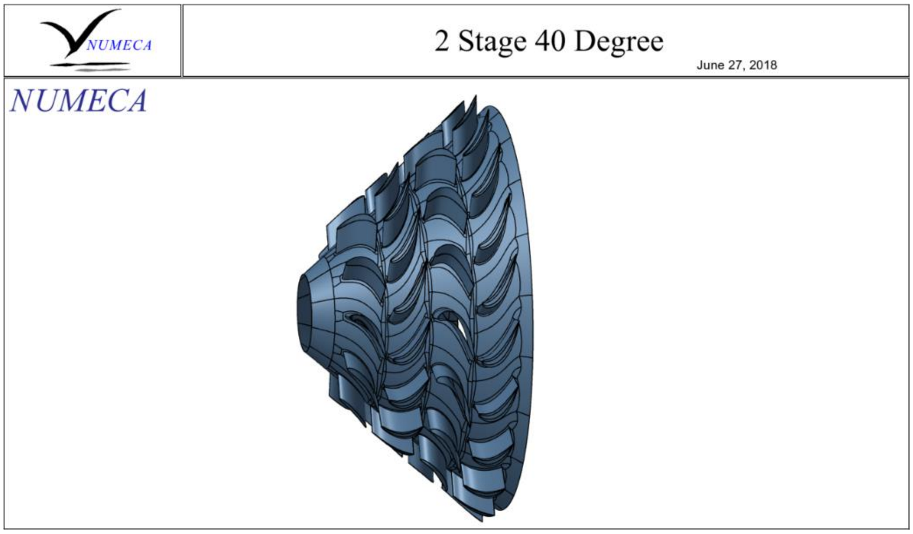

Figure 10. 2 Stage 40 Degree Turbine

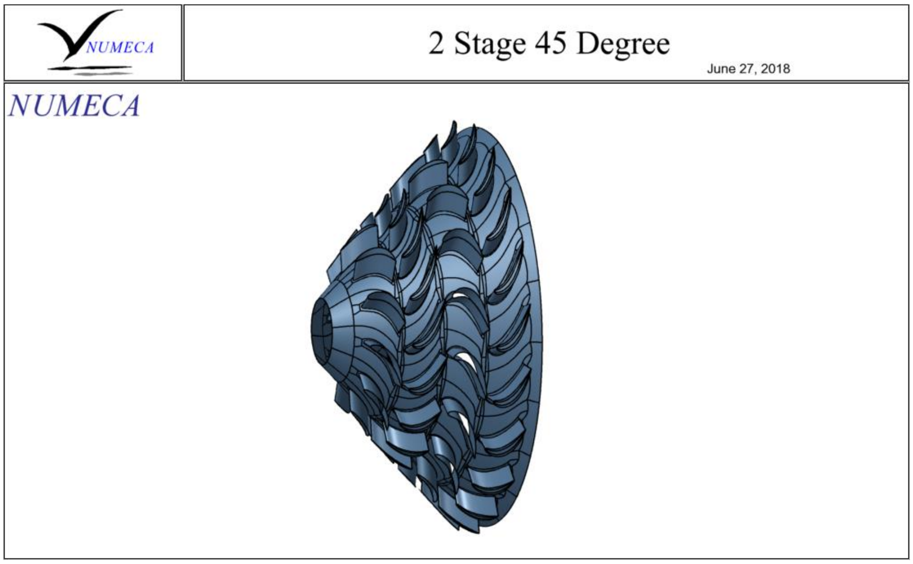

Figure 11. 2 Stage 45 Degree Turbine

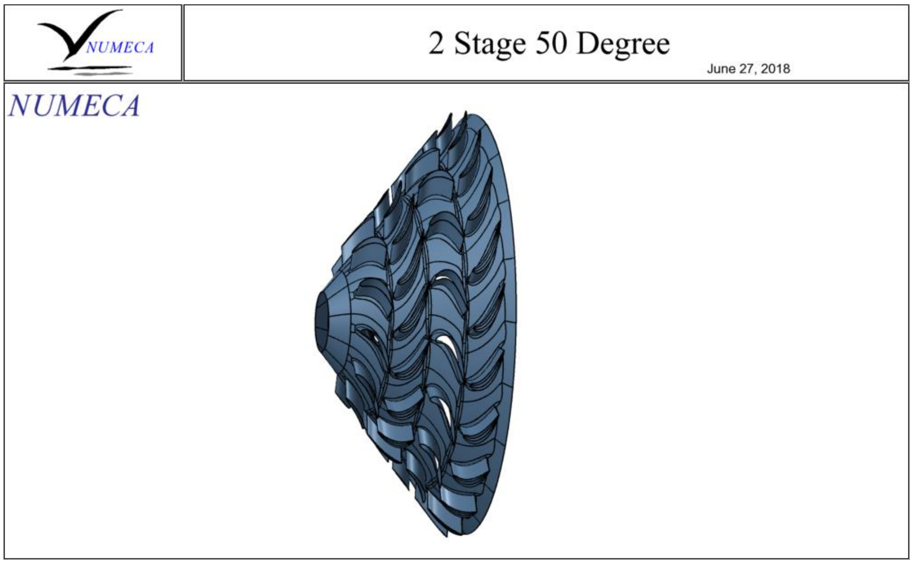

Figure 12. 2 Stage 50 Degree Turbine

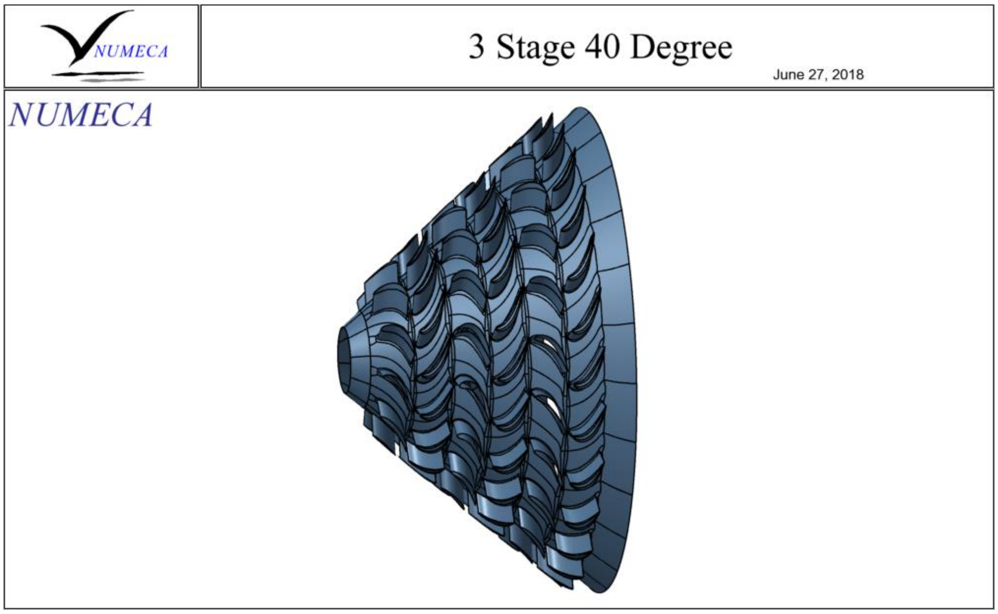

Figure 13. 3 Stage 40 Degree Turbine

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Attachment 3.
Simulation Parameter

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No.	Flow Parameters	Option	Value
1	Fluid Model	Ammonia	-
2	Flow Model	Steady	-
		Mathematical model	Turbulent Navier Stokes
		Modelling of Turbulence	Spalart Allmaras
		Reference Temperature	297 K
		Refetence Pressure	972740 Pa
3	Boundary Condition	Inlet: Total Quantities Imposed • Angle from Axial Direction (Vx extrapolated)	
		Absolut Total Pressure	972740 Pa
		Absolute Total Temperature	297 K
4		Outlet: Pressure Imposed • Averaged Static Pressure	615290 Pa
5	Numerical Model	Current grid level	1 1 1
6	Control Variables	Maximum Number of Iterations	2000
		Save Solution Every	100

Simulation Parameter

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Attachment 4.
Convergence History

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Convergence History Of Mass Flow Rate

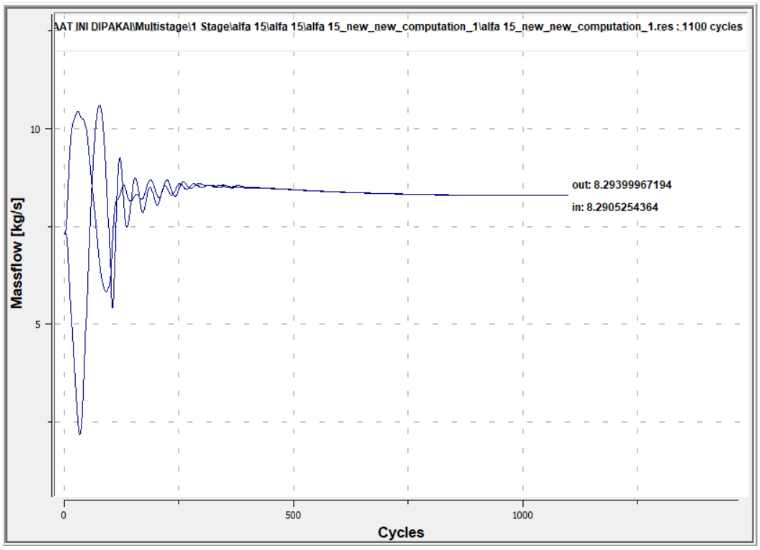

Figure 1. Single Stage

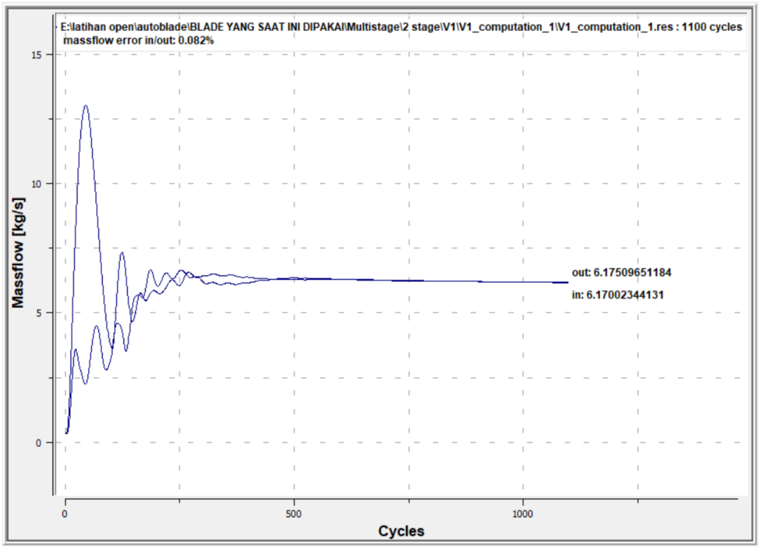

Figure 2 Stage

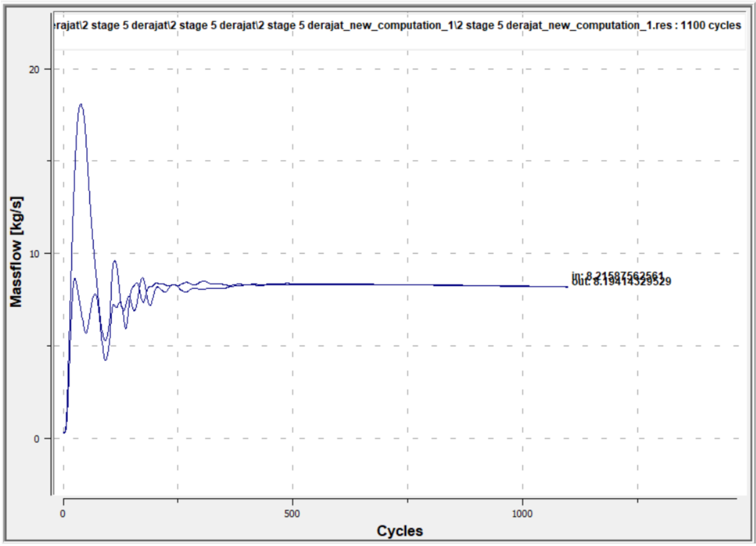

Figure 3. 2 Stage 5 Degree

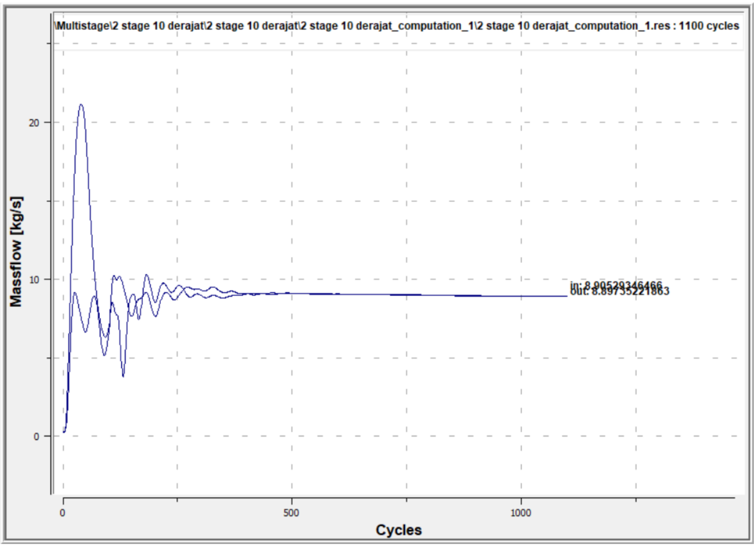

Figure 4. 2 Stage 10 Degree

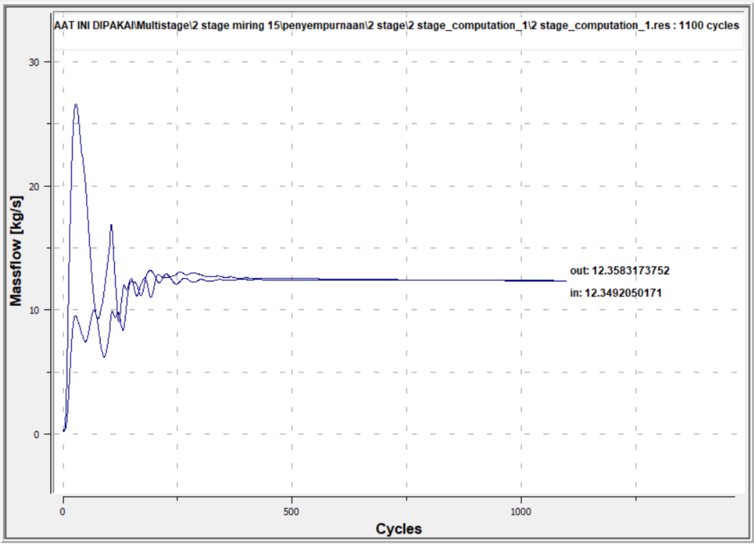

Figure 5. 2 Stage 15 Degree

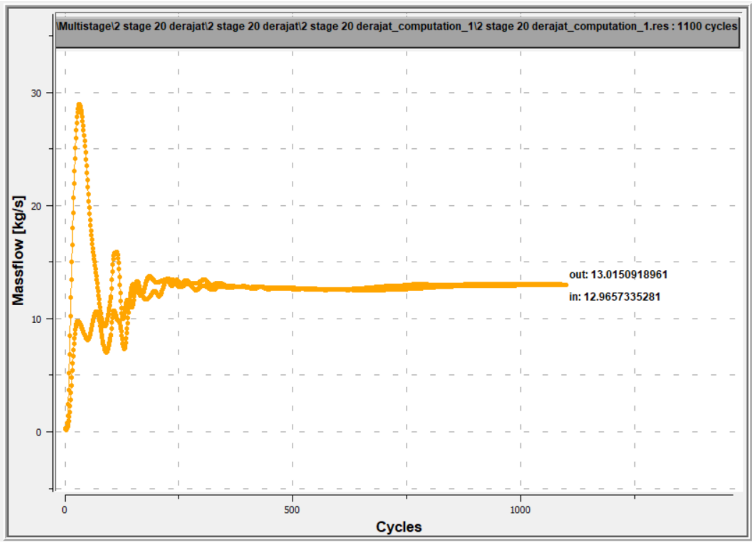

Figure 6. 2 Stage 20 Degree

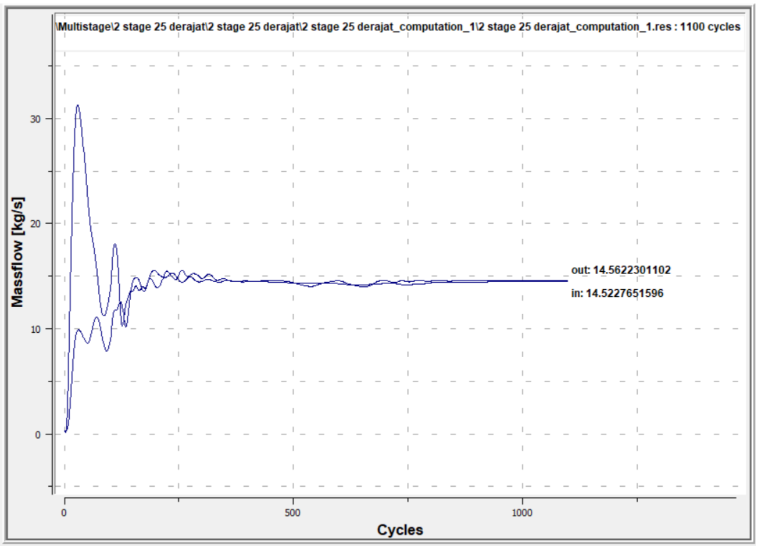

Figure 7. 2 Stage 25 Degree

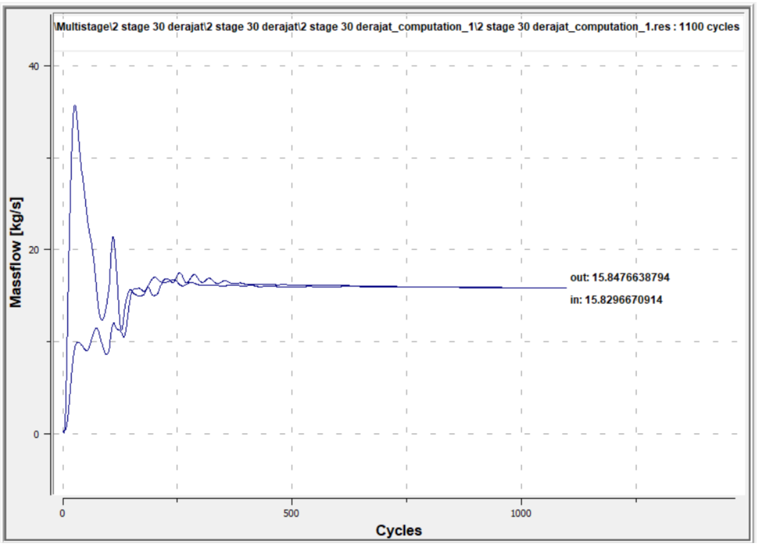

Figure 8. 2 Stage 30 Degree

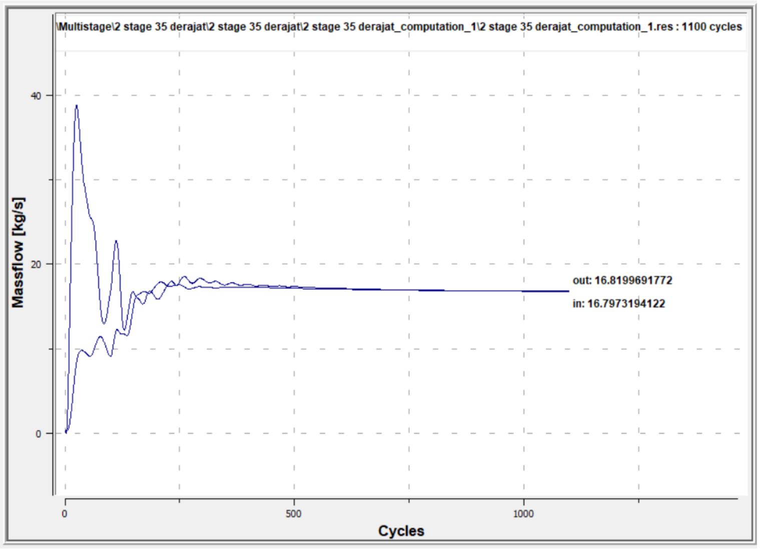

Figure 9. 2 Stage 35 Degree

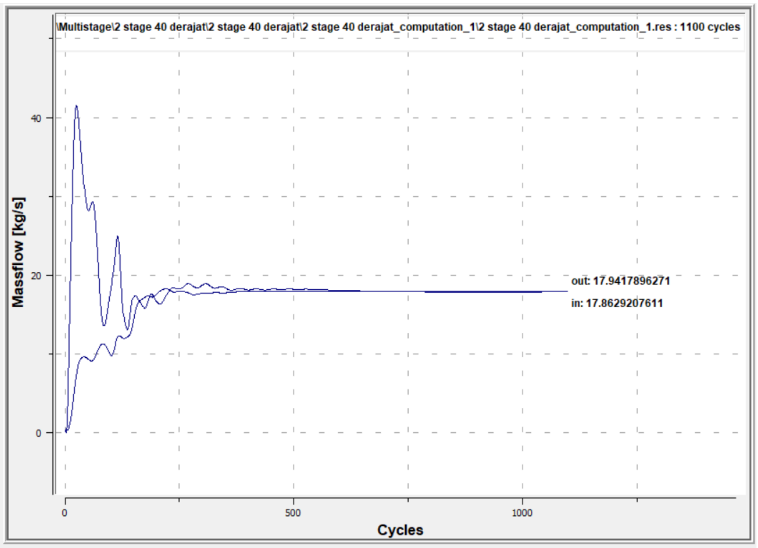

Figure 10. 2 Stage 40 Degree

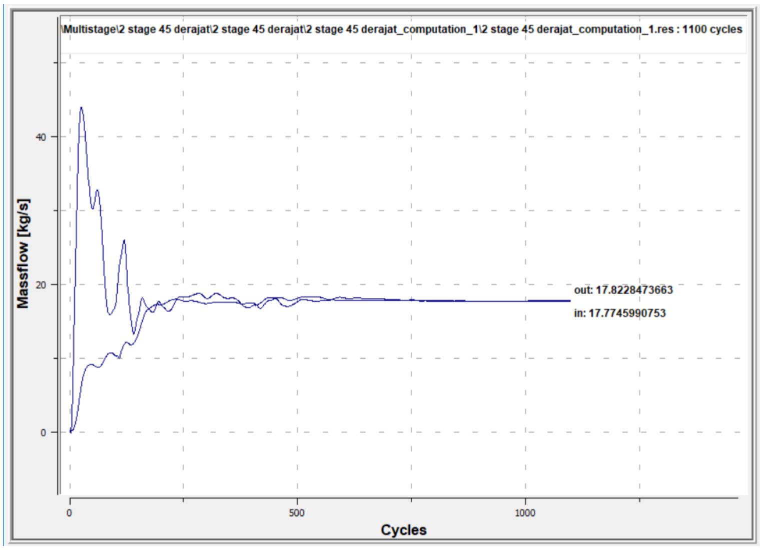

Figure 11. 2 Stage 45 Degree

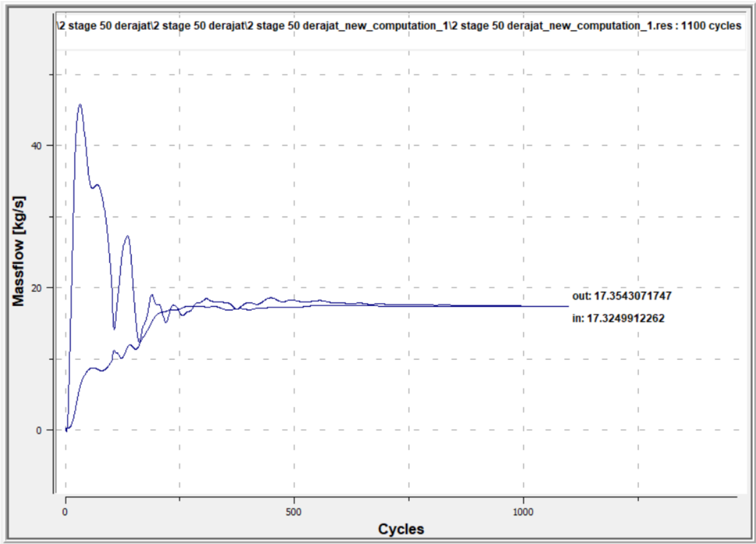

Figure 12. 2 Stage 50 Degree

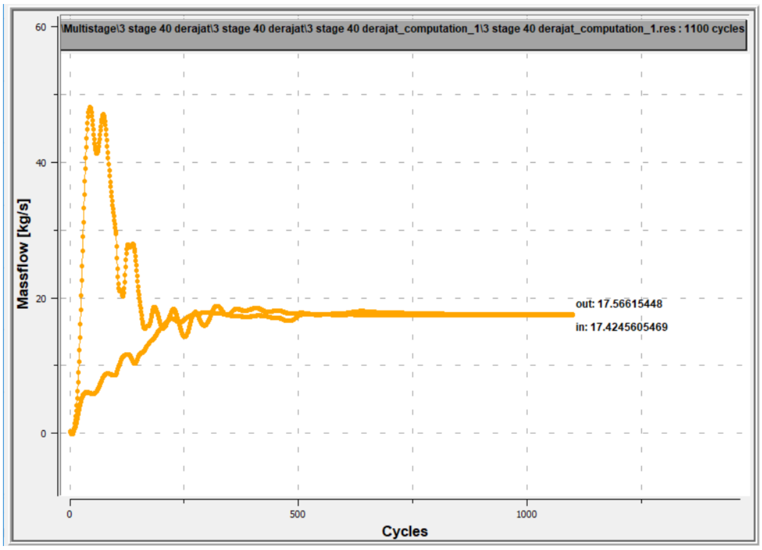

Figure 13. 3 Stage 40 Degree

Efficiency Graphic

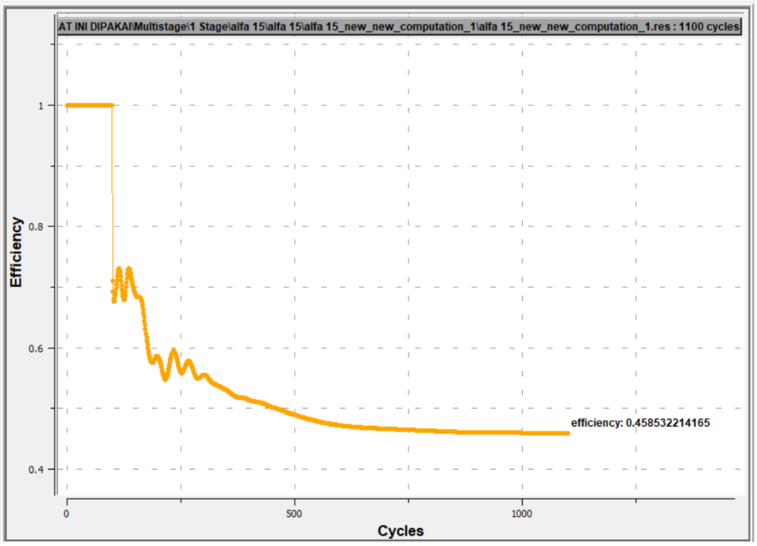

Figure 1. Single Stage

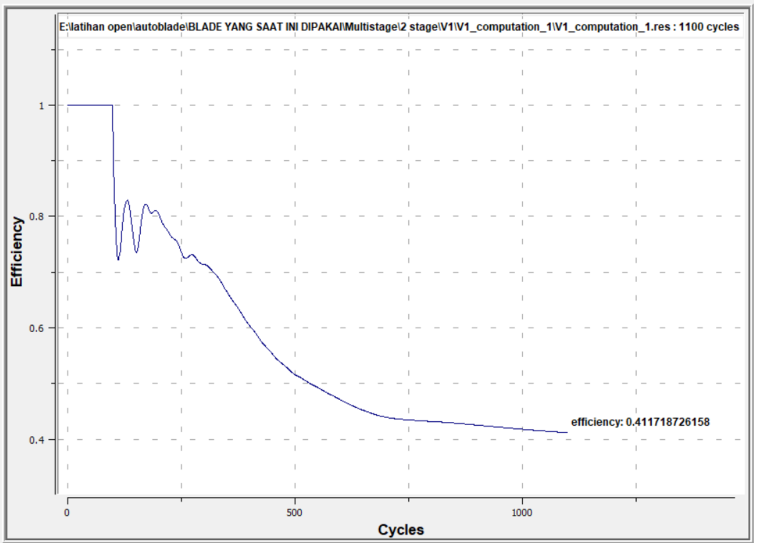

Figure 2 Stage

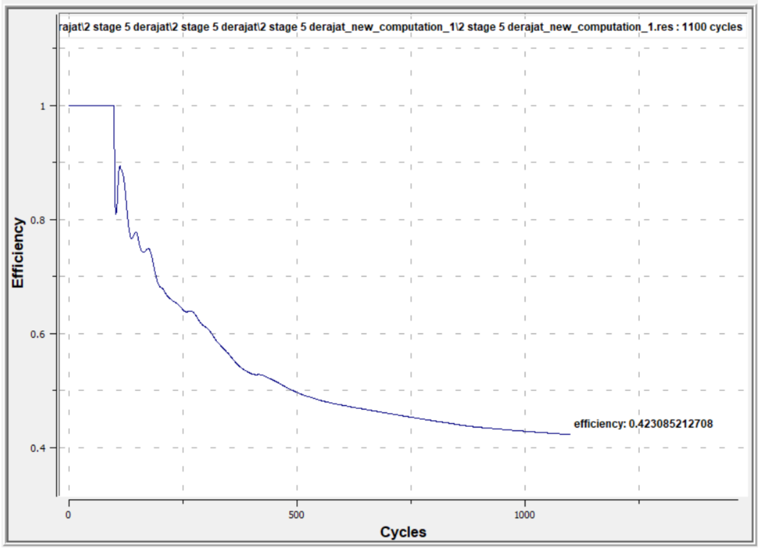

Figure 3. 2 Stage 5 Degree

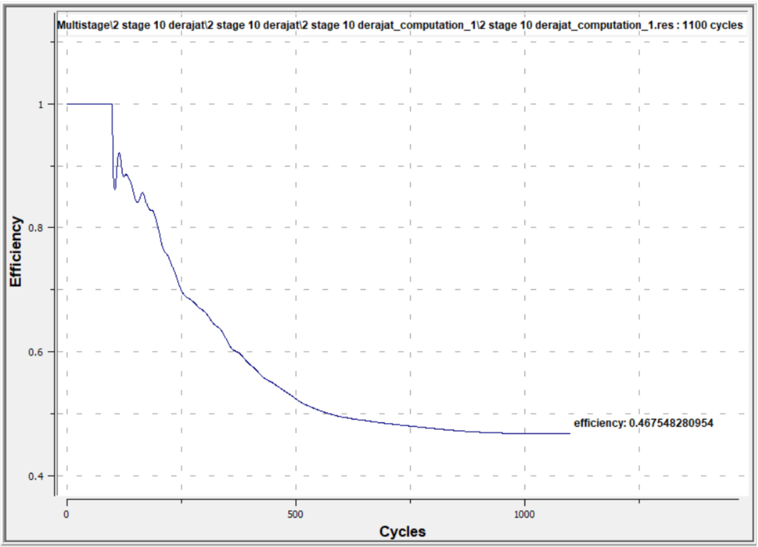

Figure 4. 2 Stage 10 Degree

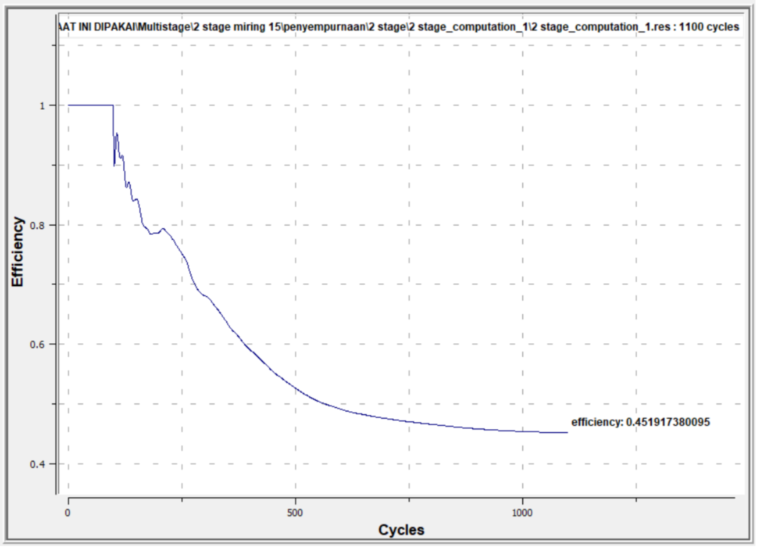

Figure 5. 2 Stage 15 Degree

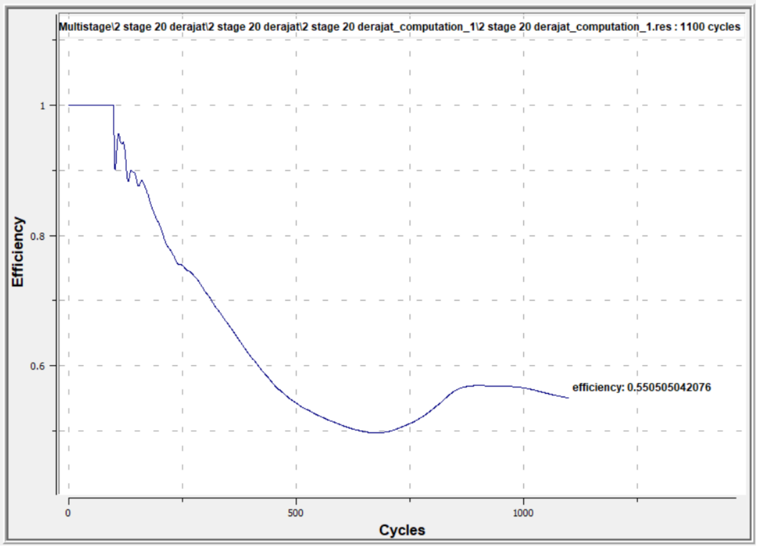

Figure 6. 2 Stage 20 Degree

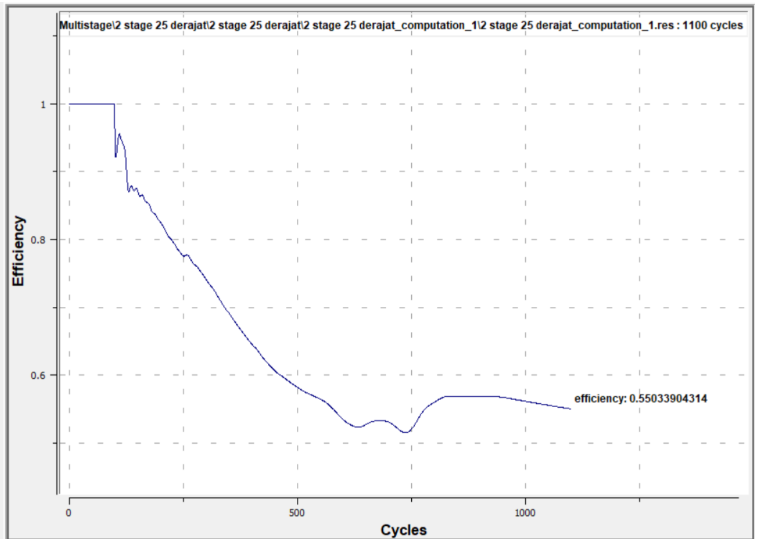

Figure 7. 2 Stage 25 Degree

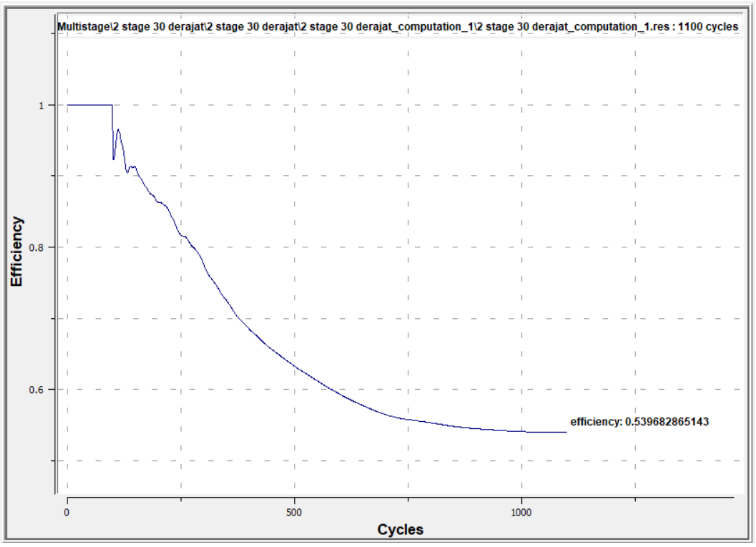

Figure 8. 2 Stage 30 Degree

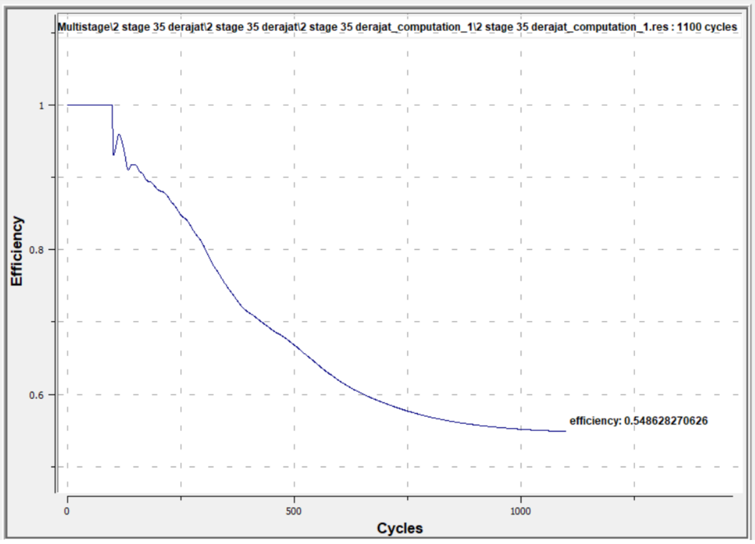

Figure 9. 2 Stage 35 Degree

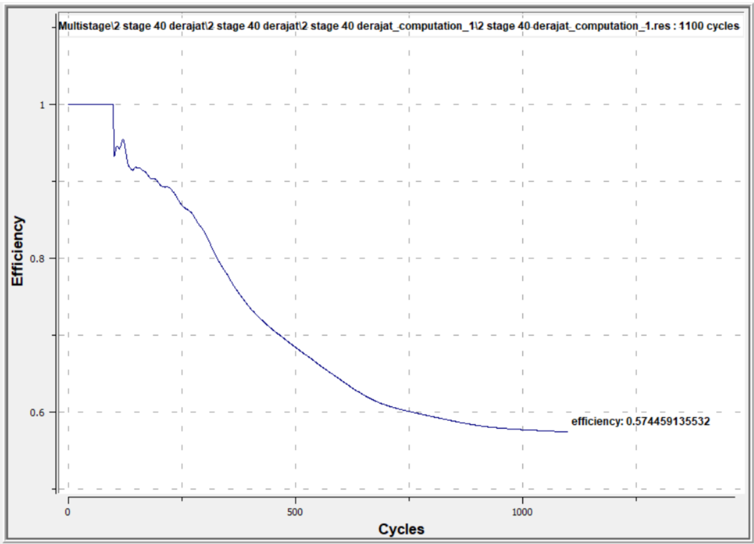

Figure 10. 2 Stage 40 Degree

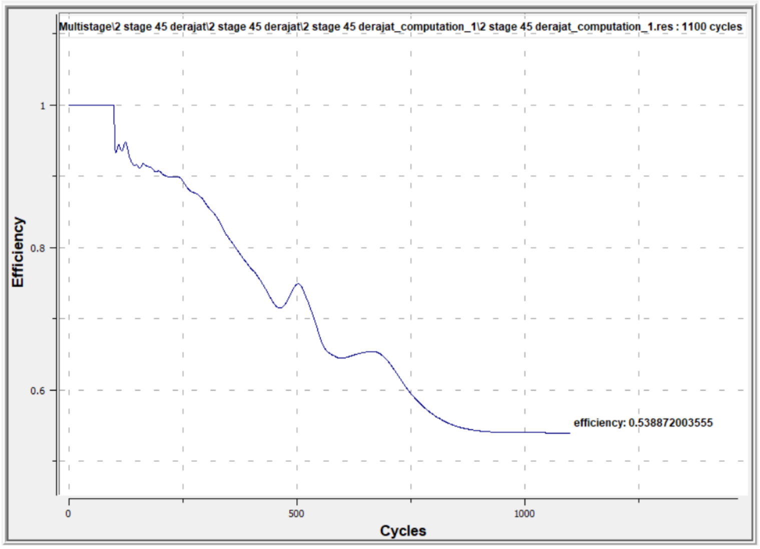

Figure 11. 2 Stage 45 Degree

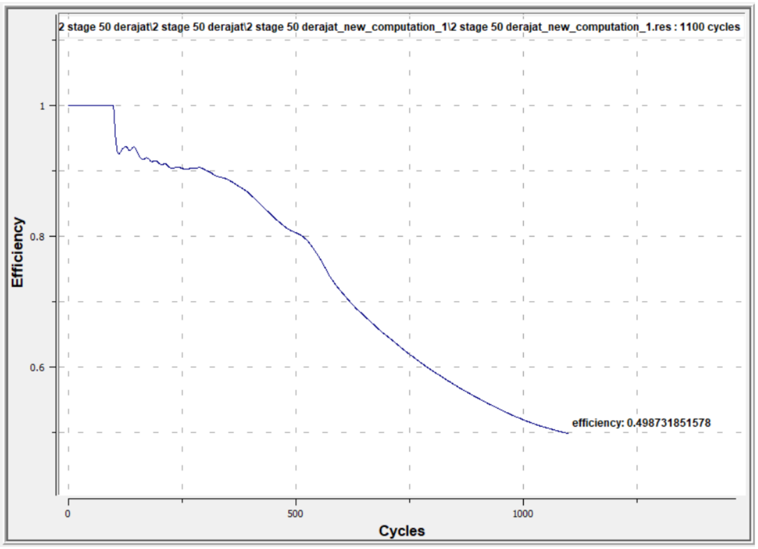

Figure 12. 2 Stage 50 Degree

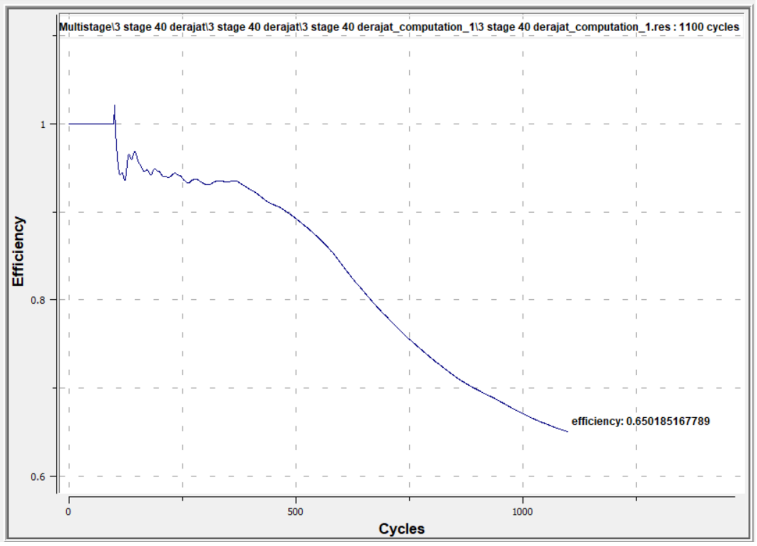

Figure 13. 3 Stage 40 Degree

Torque Graphic

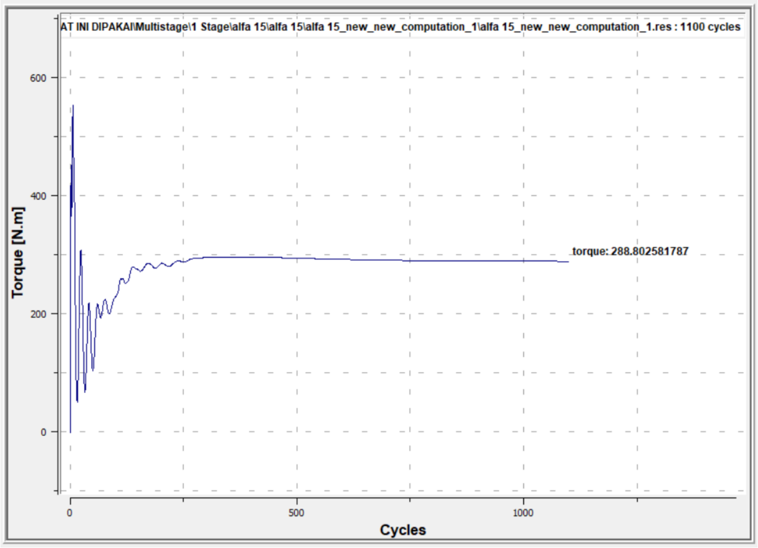

Figure 1. Single Stage

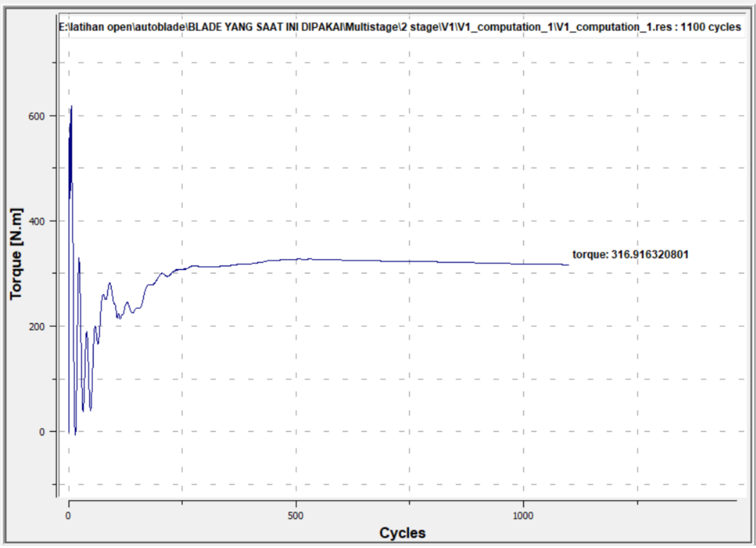

Figure 2 Stage

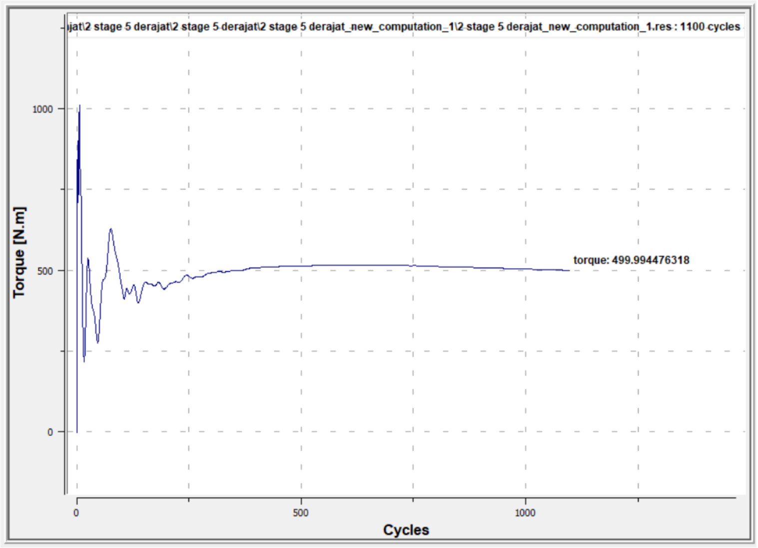

Figure 3. 2 Stage 5 Degree

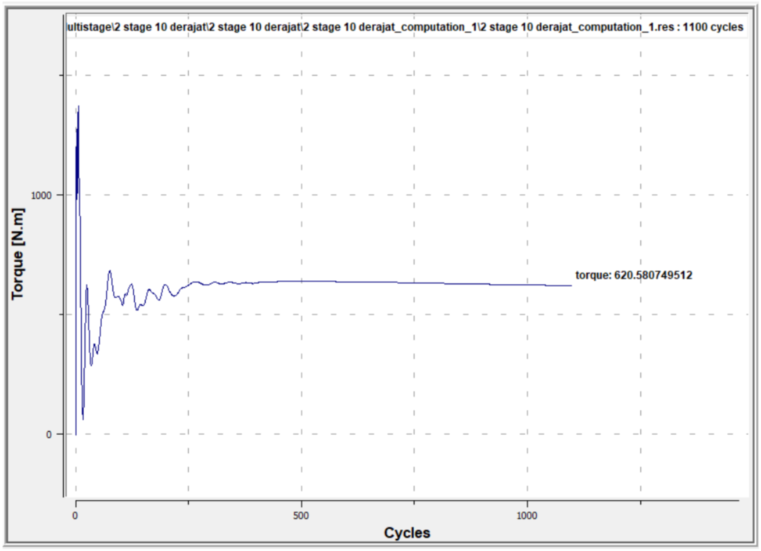

Figure 4. 2 Stage 10 Degree

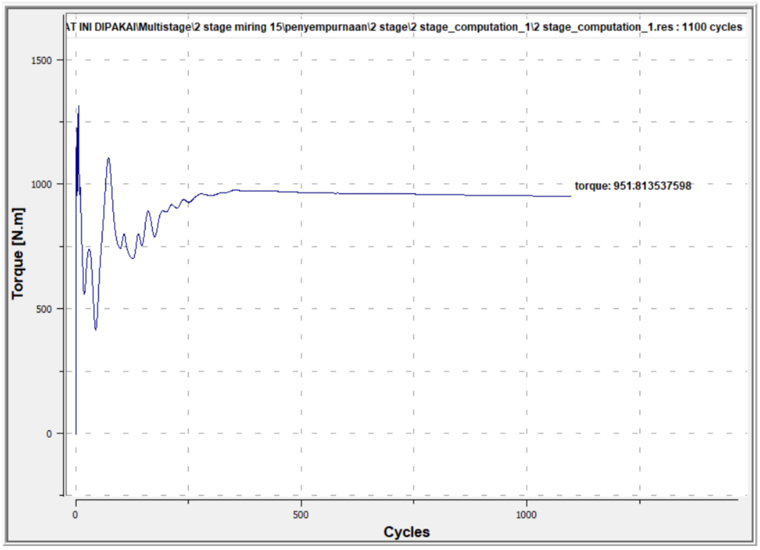

Figure 5. 2 Stage 15 Degree

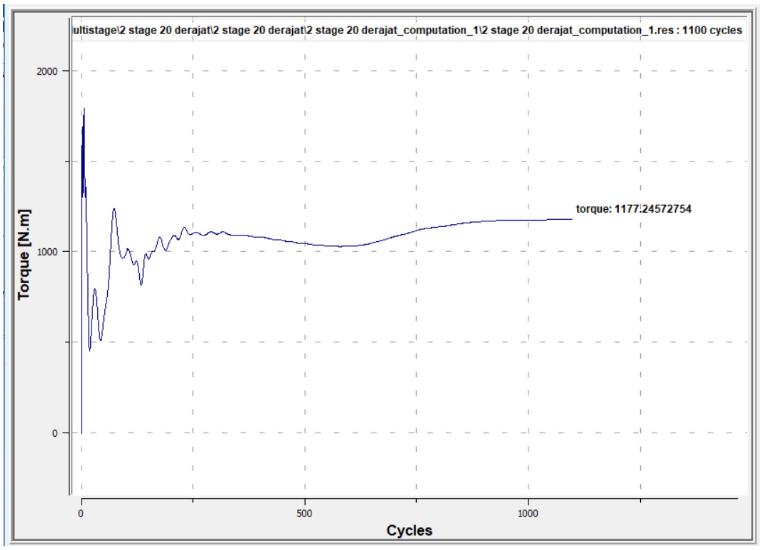

Figure 6. 2 Stage 20 Degree

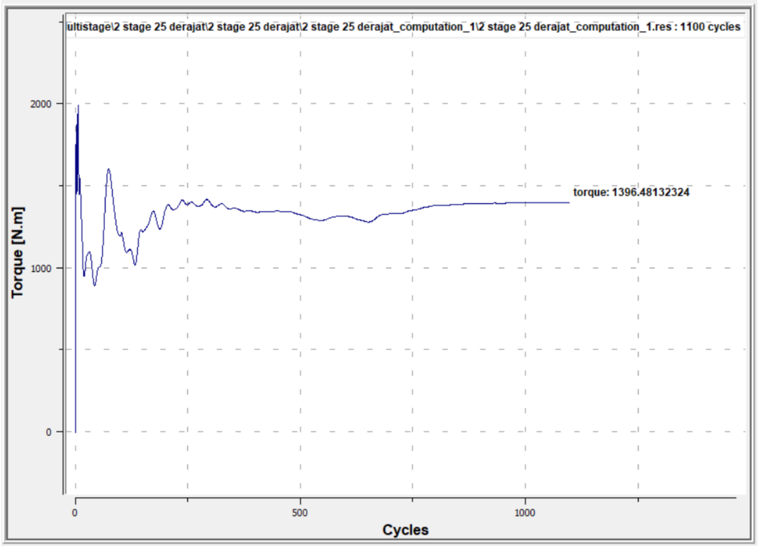

Figure 7. 2 Stage 25 Degree

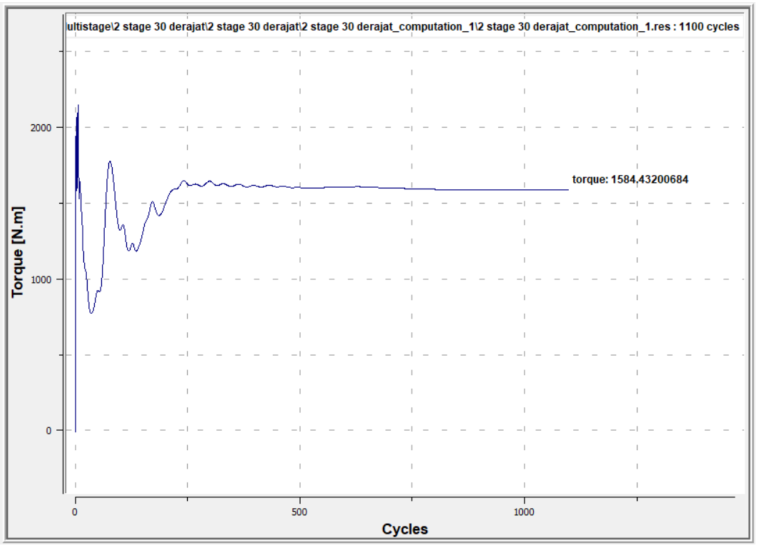

Figure 8. 2 Stage 30 Degree

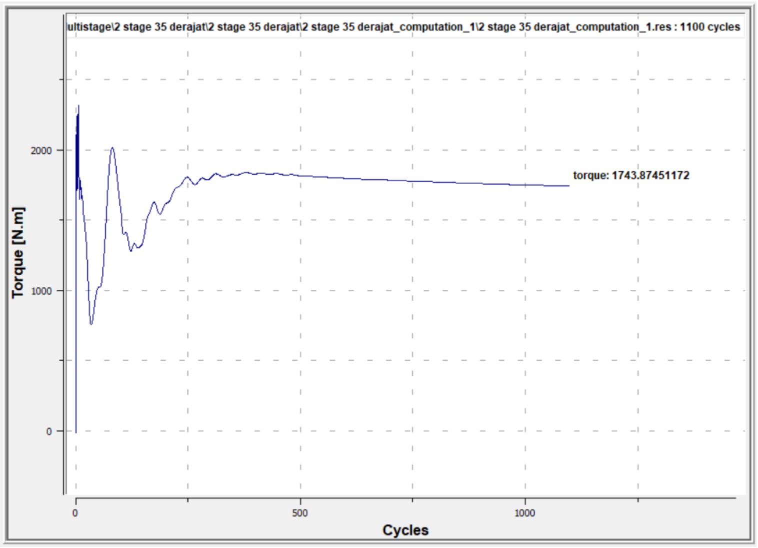

Figure 9. 2 Stage 35 Degree

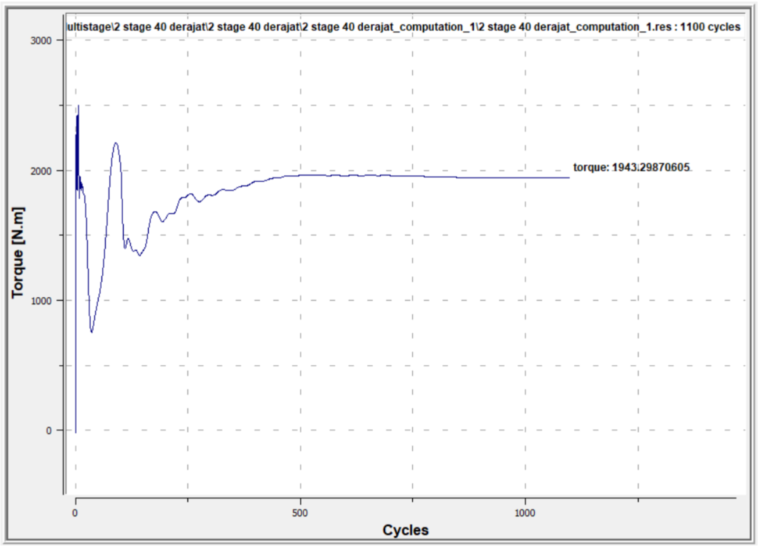

Figure 10. 2 Stage 40 Degree

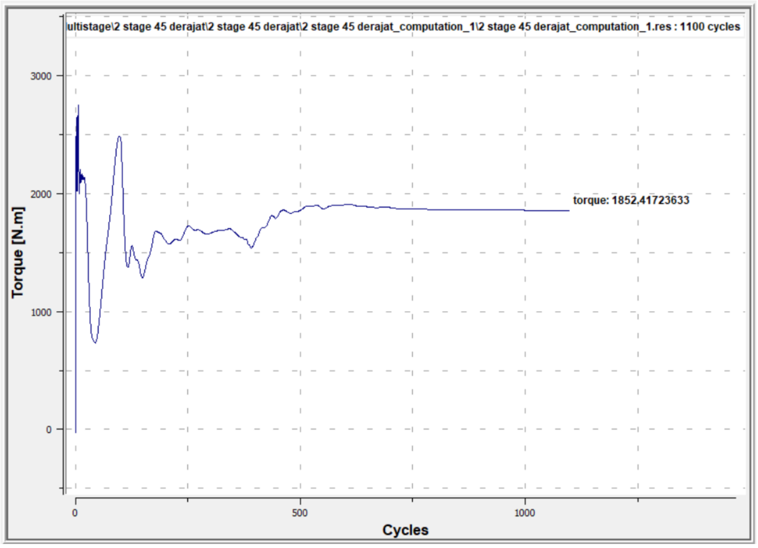

Figure 11. 2 Stage 45 Degree

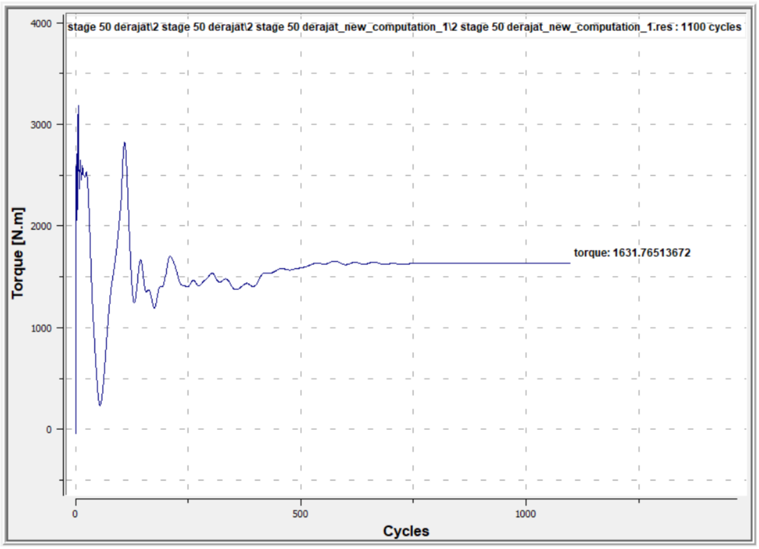

Figure 12. 2 Stage 50 Degree

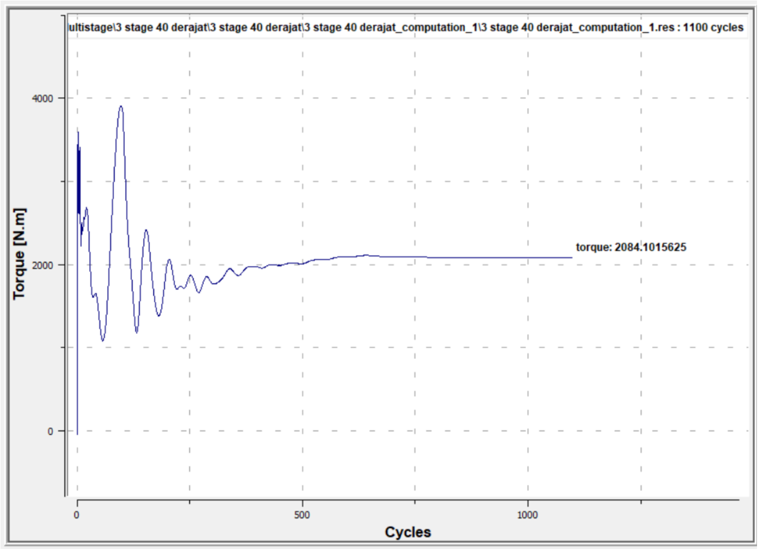

Figure 13. 3 Stage 40 Degree

BIOGRAPHY

The author’s name is Desta Rifky Aldara, handsome boy that was born on December 24th, 1995 in Kudus, Central Java. Born to be the first son derived from a couple with father named Zaenal Mutaqin, and mother named Suwanti. The author had completed the formal studies at SDN 1 Barongan (2002-2008) for elementary school, SMPN 2 Kudus (2008-2011) for junior high school, and SMAN 1 Kudus (2011-2014) for senior high school. Author proceed to pursue bachelor degree at Department of Marine Engineering (Double Degree Program with Hochschule Wismar), Faculty of Marine Technology Institut Teknologi Sepuluh Nopember Surabaya specializes in marine manufacturing and design. During the study period, the author active in several campus activities such as: entrepreneurship staff on HIMASISKAL, staff of PSDM on BEM FTK, Trainer for new student at Marine Manufacturing and Design Laboratory. For further discussion and suggestion regarding to this research, the author can be reached through email stated as below.

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Motto: Make everything simple